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TRANSACTIONS
of
AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 1047

FORTY-THIRD ANNUAL MEETING, 1937

WITH an attendance of 503 members, guests and ladies, the 43rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS was voted an outstanding success and those who journeyed to St. Louis and spent the three days, January 25 to 27, felt well repaid for their time and effort. Every one of the 22 local Chapters was officially represented and every part of the United States and Canada contributed its quota to the attendance. The attendance at technical sessions was exceptional and the discussions were lively and interesting.

The Council held an all-day meeting on Sunday, January 24, and on Monday morning, the Committee on Research and other technical committees were in session.

At 10:00 A.M. on Monday, January 25, President Larson opened the conference of Chapter Delegates and welcomed the representatives of the 22 Chapters.

W. A. Russell, chairman of the Membership Committee, opened the discussion and told of the mechanics of membership solicitation.

W. H. Driscoll, Past President of the Society, reviewed some of the important happenings in the history of the organization and mentioned some of the pioneers who had rendered outstanding service. He pointed out the value of attending the Annual and Semi-Annual Meetings of the Society and briefly reviewed the history of the first few Chapters. He commented on the relationship between the Society and the Local Chapters and the evidence of sustained interest that members have in the organization as demonstrated by the attendance of 11 past presidents at this meeting.

W. T. Jones, Past President of the Society, spoke on the duties of Chapter Presidents and told of the intangible benefits resulting from contacts made by members at Society meetings.

D. S. Boyden, First Vice-President of the Society, spoke on the subject of dues payments and the possibility of collecting Chapter dues by the Society.

He pointed out some of the advantages and disadvantages and presented two charts showing the number of enrolled Chapter Members, the average cost of operating Chapters and commented on the rate of dues that would be required to carry out a plan of this kind.

In the discussion that followed, comments were offered by E. B. Royer, H. H. Erickson, N. D. Adams, W. H. Driscoll, J. J. Hayes, F. X. Loeffler, T. H. Urdahl, G. E. Olsen, H. M. Hendrickson, James Holt and G. W. F. Myers.

The first session of the 43rd Annual Meeting of the Society was called to order by Pres. G. L. Larson, Madison, Wis., at 2:00 P.M. in the Hotel Statler, St. Louis, Mo., on Monday afternoon, January 25. C. R. Davis, president of St. Louis Chapter and chairman of the Committee on Arrangements, welcomed the members to St. Louis and recalled that the Society had met in St. Louis exactly 10 years previously.

On behalf of the Society, President Larson responded and thanked Mr. Davis and the St. Louis Chapter members for their cordial greeting.

President Larson then read the annual report of the president.

Report of the President

It is the usual practice for the President of this Society to come before the members at the time of the Annual Meeting and give them an account of his stewardship during the year just passed. In addition to a report on general activities of the Society he may take advantage of his opportunity and include such general comments as he may feel inclined to give on matters of broader interest pertaining to the future welfare of the organization.

In this brief report, I shall refer only briefly to the status of the major activities of the Society and will confine my remarks more particularly to some of the outstanding problems of the Society as they appear to me. More detailed accounts of Society activities will be brought out in the Report of the Council, the Report of the Secretary, and the Reports of the Research, Guide, Finance, and other Committees.

During my term of office I have visited all but one of our twenty-two Chapters located from coast to coast and in the Dominion of Canada. I have also represented the Society as a speaker at meetings of such organizations as the *Air Conditioning Society of San Francisco*, the *Seattle Engineers Club*, the *Illinios Fuel Merchants Association*, the *Engineers Society of Milwaukee*, and a number of service clubs. The new Texas Chapter is the only Chapter of the Society that your President has not had the pleasure of visiting.

Past experience as chairman of some of our major committees and the fair cross-section of the views of our members obtained on my visits to Chapters has impressed upon me certain ideas that I want to leave with you for any value that they may have.

Before touching upon these ideas let me first make a very brief reference to the condition of the Society in some of its major phases.

Membership. The number of members enrolled at the present time is the largest in the history of the Society. Four hundred and ninety applications were filed during the calendar year of 1936. Due to the number of new members the income from dues was greater than the amount anticipated in the budget for 1936.

The Guide. THE GUIDE 1936 was distributed in greater quantity than any previous edition—14,500 copies to date. THE GUIDE advertising showed an increase of 50 pages over last year. This publication is not only rapidly bringing increased prestige to our Society but it is also becoming one of our major sources of income.

Research. The research division of our organization has passed through some trying experiences in recent years, but I am happy to report that the skies are rapidly clearing and activities are being resumed that had to be dropped through lack of funds. Among other things we are once more getting into position where we

can pay our way in connection with co-operative research agreements with the colleges and the current year should see increased activity in this direction.

In closing this report I wish to refer briefly to a number of situations which, for the future welfare of the Society, need serious consideration.

THE GUIDE has become one of the major sources of income for the Society and it should be able to pay its way better than it has in the past. You may be surprised to learn that the members of the Guide Publication Committee have not only devoted many hours of their valuable time in getting out this most valuable piece of work, but in addition they have paid their own expenses to meetings of that Committee.

In view of the income that THE GUIDE brings to the Society the very least we can do is to have the Council vote to defray the expense of the Guide Publication Committee to meetings of that group.

I had the experience of being chairman of the Committee on Research for two years, and I began that career by being elected chairman without having had any previous experience on the Committee. Colonel Danielson has just passed through the same experience this year. Electing to the chairmanship a man who has had no previous experience on the Committee is a decided mistake. It is very unfair to the individual himself and is not conducive to the best interests of the Society. I remember that I spent a large part of my first year as chairman of the Committee on Research trying to find out what it was all about and Colonel Danielson has had a similar experience this past year.

I believe that a ruling should be made that no one can be elected to the chairmanship of the Committee on Research until he has had at least one year's experience on that committee. I think that such a rule would be a good rule for all of our regular committees.

And while I am on the subject of Research I wish to advocate a closer relationship between our research division and our main administrative office in New York, particularly in relation to publicity and the solicitation of funds for research.

Under the present arrangement each new chairman has to appoint his own Research Finance Committee and develop his own publicity campaign. The year is nearly over by the time these are organized and begin to function. The Research Chairman is lucky if he can induce someone to undertake the job of chairmanship of the Research Finance Committee. Whoever undertakes it is new at the game and the year is practically over before much has been accomplished. The next year and the following year new groups go through the same inefficient and painful experience. It seems to me that Research Fund Solicitation and Research publicity can be much more efficiently handled by additions to the staff in our New York office, specifically trained for that purpose. The chairman can then be relieved to carry on the main purpose of the job, the development and advancement of research projects.

Respectfully submitted,
G. L. LARSON, *President.*

Report of the Treasurer

A. J. Offner, New York, treasurer of the Society, reported that the finances were in excellent condition and called attention to a substantial increase in the General Fund as compared with the previous year. He explained that the detailed report of Society operations would be given by the chairman of the Finance Committee.

Report of the Finance Committee

J. F. McIntire, Detroit, chairman of the Finance Committee, presented the report of the Society's certified public accountant showing the condition of the finances for the calendar year 1936. He pointed out that income had exceeded budget expectations and that expenditures had been kept within the amount allotted, so that a very satisfactory condition of Society affairs could be presented.

XUM

Report of Certified Public Accountant

January 18, 1937.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS,
51 MADISON AVE.,
NEW YORK, N. Y.

Gentlemen:

Pursuant to your request we made an examination of the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y., and the RESEARCH FUND for the calendar year ended December 31, 1936, and submit herewith our report.

The work covered a verification of the Assets and Liabilities as of the date previously stated and a review of the operating accounts for the year ended December 31, 1936. For this period recorded cash receipts were traced into the depositories; the cancelled bank vouchers were inspected and compared with the cash records and the Dues Income was accounted for.

Submitted herewith is a Balance Sheet showing the financial condition of the Society on December 31, 1936, and your attention is directed to the following comments thereon:

CASH

Cash on Deposit was verified by direct communication with the banks and reconcilement of the amounts reported to us with the balance shown by the books of the Society.

The Petty Cash kept in the New York Office was verified by count.

A summary of the cash accounts follows:

				Research	
ACTIVE	GENERAL FUND	ENDOWMENT FUND	F. PAUL ANDERSON MEDAL FUND	ENDOWMENT FUND	UNRESTRICTED FUNDS
Bankers Trust Co.	$13,390.39	$4,985.23	....	....	$11,598.32
Chase National Bank	2,506.39	....	....	....	....
Forbes National Bank, Pittsburgh, Pa.	....	....	....	....	446.29
Bowery Savings Bank	8,131.26	....	....	....	....
Central Savings Bank	7,261.19	....	....	....	....
Dry Dock Savings Institution	5,607.86	....	....	....	....
Emigrant Industrial Savings Bank	7,782.69	....	....	....	....
Bank for Savings	....	....	$1,108.27	$451.99	2,965.55
Provident Institution for Savings, Boston, Mass.	3,482.33	....	....	....	....
	$48,162.11	$4,985.23	$1,108.27	$451.99	$15,010.16
On Hand	100.00	....	....	....	50.00
	$48,262.11	$4,985.23	$1,108.27	$451.99	$15,060.16
FROZEN	$ 500.00	....	$ 33.22	$213.53	.....

MARKETABLE SECURITIES

There is attached hereto a schedule of negotiable bonds which were verified by direct communication with the Bankers Trust Company where same are deposited for safekeeping. No adjustment has been made of the $2,616.36 shrinkage in the market value of these securities. These have been included in the attached Balance Sheet at cost.

ACCOUNTS RECEIVABLE

A list of the Membership Dues Receivable as of December 31, 1936, furnished to us by the management was checked to the individual ledger cards and found in agreement with the General Ledger Control. The unpaid dues may be summarized as follows:

Dues billed in 1936	$ 6,141.71
Dues billed in 1935	1,880.00
Dues billed in prior years	6,456.56
TOTAL	$14,478.27

Amounts due from GUIDE Advertisers and Other Accounts Receivable verified by trial balance of the individual ledger accounts were found in agreement with the General Ledger Control.

The sum of $2,066.19 has been added to the Reserve for Dues Doubtful of Collection, making a total Reserve of $9,417.19. Uncollectable Guide Advertisements and Other Receivables amounting to $535.88 have been written off, leaving a balance of $1,964.12. It is our opinion that the Reserves now on the books are sufficient to cover any further losses from realization.

INVENTORIES

Items appearing under this caption are based upon quantities submitted by the management and computations made by us. The Transactions Inventory priced at cost may be summarized as follows:

Prior Years	$1,811.06
1931	404.67
1932	433.29
1933–34	722.09
TOTAL	$3,371.11

FURNITURE, FIXTURES AND LIBRARY

Furniture, Fixtures and Library are shown herein at the book values without appraisal by us; we did, however, provide for depreciation of Furniture and Fixtures at the rate of ten per cent (10%) per annum.

ACCOUNTS PAYABLE

On December 31, 1936, there remained unpaid invoices amounting to $446.96, also the sum of $10,960.00 estimated by the management necessary to complete and make the first mailing of THE 1937 GUIDE.

PAYABLE TO RESEARCH FUND

The amounts due the Research Fund representing 40 per cent of Members and Associates dues were determined from computations made by us in accordance with Section 5, Article 3, of the By-Laws.

ACCRUED ACCOUNTS

Additional compensation to the Secretary and the clerical staff of the Society has been computed in accordance with the instructions of the Finance Committee.

RESERVE FOR PUBLICATIONS

There has been reserved the sums of $3,000.00 and $3,500.00 respectively for publication of Volumes 41 and 42 of Transactions. These provisions have been made in accordance with Council Authorization.

DEFERRED INCOME

Members Prepaid Dues were ascertained by trial balance of the individual ledger cards. Prepaid Dues from Proposed Members were verified from inspection of the applications found on file.

RESERVE FOR ENDOWMENT FUND ADJUSTMENT

The original appropriation of the Endowment Fund of $19,641.18 shows an actual reduction in principal of $3,277.90 due to losses incurred from the Sales of Securities. In addition there exists a shrinkage in market value of securities now in portfolio of this Fund amounting to $2,796.05. The Council in order to replenish the Fund to its original amount and to provide for the losses aforementioned at a meeting held on January 27, 1936, has authorized the setting aside, out of 1935 income, the sum of $7,500.00.

GENERAL FUND

There is atached hereto a schedule showing the changes that have occurred in the General Fund during the calendar year 1936.

SPECIFIC FUND

The specific funds are shown on the attached Balance Sheet after adjustment for interest earned and appropriations authorized by Council.

Operations of the Research Funds for the calendar year 1936 are detailed in schedules attached hereto.

OPERATIONS

There is attached hereto Statements of Income and Expenses for the calendar year 1936 showing a net income of $3,445.98 from Society activities and $17,182.98 from GUIDE Operations. In the preparation of these statements 30 per cent of the Salaries and Office Overhead expenses have been allocated to Guide Operations in the attached statements.

SPECIAL APPROPRIATIONS

The appropriation of $3,000.00 for the New York Academy of Medicine Cooperative Project has not been set up on the books by us in view of the fact that the required outside contributions have not been received.

Respectfully submitted,
PUBLIC AUDIT COMPANY,
(Signed) FRANK G. TUSA,
CERTIFIED PUBLIC ACCOUNTANT.

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.
December 31, 1936

ASSETS

SOCIETY				
CASH				
On Deposit		$48,162.11		
On Hand		100.00		
In Closed Bank		500.00	$48,762.11	
INVESTMENTS (AT COST)				
Securities (Market Value $11,705.00)		11,525.31		
ADD: Accrued Interest		61.46	11,586.77	
ACCOUNTS RECEIVABLE				
Membership Dues	$14,478.27			
LESS: Reserve for Doubtful	9,417.19	5,061.08		
Advertisers and Sundry Debtors	23,328.31			
LESS: Reserve for Doubtful	1,964.12	21,364.19	26,425.27	
INVENTORIES				
Transactions		3,371.11		
Emblems and Certificate Frames		77.05		
Postage		360.72	3,808.88	
PERMANENT				
Library		300.00		
Furniture and Fixtures	5,865.21			
LESS: Reserve for Depreciation	4,879.37	985.84	1,285.84	
DEFERRED CHARGES				
Chapter Meeting Allowance			500.00	$92,368.87
SPECIFIC FUNDS				
ENDOWMENT FUND				
Cash on Deposit			4,985.23	
Securities at Cost (Market Value $9,122.60)		11,918.65		
ADD: Accrued Interest		111.35	12,030.00	17,015.23
F. PAUL ANDERSON AWARD FUND				
Cash on Deposit			1,108.27	
Cash in Closed Bank			33.22	1,141.49
RESEARCH FUND				
ENDOWMENT FUND				
Cash on Deposit		451.99		
Cash in Closed Bank		213.53	665.52	
UNRESTRICTED FUNDS				
Cash on Deposit		13,154.82		
Cash on Hand		50.00		
Cash on Hand for Deposit		1,855.34	15,060.16	15,725.68
				$126,251.27

LIABILITIES AND CAPITAL

SOCIETY			
ACCOUNTS PAYABLE		$11,406.96	
PAYABLE TO RESEARCH FUND			
On Dues as and when Collected		2,397.53	
ACCRUED ACCOUNTS			
Compensation—Secretary and Staff		2,250.41	
RESERVE FOR PUBLICATIONS			
Transactions Vol. 41 and 42		6,500.00	
DEFERRED INCOME			
Prepaid Dues—Members	$1,109.11		
Prepaid Dues—Proposed Members	1,149.68	2,258.79	
RESERVE FOR ENDOWMENT FUND REPLENISHMENT		7,500.00	
TOTAL LIABILITIES		32,313.69	
GENERAL FUND			
Society		60,055.18	$92,368.87
SPECIFIC FUNDS			
ENDOWMENT FUND			
Principal		16,363.28	
Unexpended Income		651.95	17,015.23
F. PAUL ANDERSON AWARD FUND			
Principal		1,000.00	
Unexpected Income		141.49	1,141.49
RESEARCH FUND			
ENDOWMENT FUND			
Principal	600.00		
Unexpended Income	65.52	665.52	
UNRESTRICTED FUND			
Balance—January 1, 1936	8,178.62		
Excess of Income over Expenses for the Year Ended December 31, 1936	6,881.54	15,060.16	15,725.68
			$126,251.27

NOTE "A"—This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

STATEMENT OF INCOME AND EXPENSES

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.
For the Calendar Year Ended December 31, 1936

INCOME

FROM MEMBERS				
DUES—RENEWALS				
Members and Associates			$30,186.00	
LESS: Cancellations		$2,394.00		
Provision for Uncollectable		2,066.19	4,460.19	
			25,725.81	
LESS: 40% to Research Laboratory			11,116.80	
			14,609.01	
Junior and Student Members			2,595.50	$17,204.51
DUES—NEW MEMBERS				
Members and Associates	3,823.86			
LESS: 40% to Research Laboratory	1,529.54		2,294.32	
Junior and Student Members			1,068.56	3,362.88
				20,567.39
INCOME FROM SALES OF EMBLEMS AND CERTIFICATE FRAMES				77.27
TOTAL INCOME FROM MEMBERS				$20,644.66
FROM OTHER SOURCES				
Editorial Contract			12,858.37	
LESS: Members' Subscriptions			3,155.41	9,702.96
Net Income—Reprints, Books and Codes				269.63
Net Income—Transactions—Per Schedule				1,636.06
TOTAL INCOME FROM OTHER SOURCES				11,608.65
FROM INVESTMENTS				
Interest from Current Funds				603.80
Interest from Securities				305.04
TOTAL INCOME FROM ALL SOURCES				$33,162.15

EXPENSES

APPORTIONABLE EXPENSES			
Salaries—Secretary and Staff	$17,654.44		
Traveling—Secretary and Staff	1,336.52		
Rent and Light	2,598.78		
Telephone	524.72		
Telegrams	277.73		
Postage	1,836.34		
Depreciation—Furniture and Fixtures	564.95		
General Printing	788.90		
Office Supplies	386.70		
Multigraphing and Stencils	345.57		
Professional Services	400.00		
Bank Charges	57.05		
Miscellaneous	606.10		
	$27,377.80		
LESS: 30% Applicable to Guide Operations	8,213.34	$19,164.46	
		$19,164.46	$33,162.15
OTHER EXPENSES			
Meetings—Annual and Semi-Annual	$1,409.75		
President's Fund	1,292.96		
Yearbook 1936	37.38		
Yearbook 1937	939.19		
Awards, Membership Certificates, Medals, etc	416.37		
Exchange Service Plan	81.63	4,177.28	
OTHER DEDUCTIONS			
Chapter Meetings Allowance	1,000.00		
Provision for 1936 Transactions	3,500.00		
Special Appropriations	1,824.43		
Contribution to American Standards Association	50.00	6,374.43	29,716.17
NET INCOME FROM SOCIETY ACTIVITIES			$3,445.98

BUDGET COMPARISON

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

For the Calendar Year Ended December 31, 1936

INCOME

	Actual	*Budget Provision*	*Increases Decreases*
Membership Dues	$20,567.39	$20,000.00	$ 567.39
Profit—Emblem and Certificate Frames Sales	77.27	100.00	22.73
Journal Reprints, Books, etc	304.19	250.00	54.19
Transactions	1,636.06	750.00	886.06
Editorial Contract	12,858.37	12,800.00	58.37
Interest on Reserve and Current Funds	908.84	600.00	308.84
Guide Advertising and Copy Sales	56,525.49	48,000.00	8,525.49
	$92,877.61	$82,500.00	$10,377.61

EXPENSES

COST OF GUIDE			
Printing and Binding	$15,700.00	$16,500.00	$ 800.00
Mailing Expense	1,400.00	3,500.00	2,100.00
Paper Cost	3,955.02	3,500.00	455.02
Cuts, Engraving, etc	642.18	500.00	142.18
Advertising Sales Cost	3,184.20	2,500.00	684.20
Copy Sales Cost	3,248.16	5,000.00	1,751.84
Editorial Service	3,000.00	3,000.00	....
	$31,129.56	$34,500.00	$ 3,370.44
GENERAL SOCIETY ACTIVITIES			
Contribution American Standards Association	$ 50.00	$ 50.00	....
Secretary and Staff Salaries	17,654.44	16,500.00	$ 1,154.44
Printing Codes	34.56	250.00	215.44
Subscriptions	3,155.41	5,000.00	1,844.59
General Printing	788.90	500.00	288.90
Postage	1,836.34	2,500.00	663.66
Meetings	1,409.75	1,500.00	90.25
Chapter Allowance	1,000.00	1,000.00	....
Travel—Secretary and Staff	1,336.52	1,000.00	336.52
President's Fund	1,292.96	1,500.00	207.04
Rent and Light	2,598.78	2,700.00	101.22
General Office Expense	687.73	600.00	87.73
Office Supplies	386.70	500.00	113.30
Furniture and Fixtures	564.95	200.00	364.95
Telephone	524.72	500.00	24.72
Telegraph	277.73	300.00	22.27
Multigraphing	345.57	400.00	54.43
Professional Services	400.00	500.00	100.00

Awards, Members' Certificates and Medals	416.37	350.00	66.37
Bank Charges	57.05	350.00	292.95
Yearbook	976.57	1,000.00	23.43
Transactions—1935	3,500.00	3,500.00	...
Special Appropriations	1,824.43	5,000.00	3,175.57
	$72,249.04	$80,200.00	$ 7,950.96

Report of Membership Committee

The report of the Membership Committee was given by W. A. Russell, chairman, Kansas City, who said that the membership increase for the year totaled 469 members and that the present status of Society membership was at an all time high.

Honorary Members	3
Presidential Members	25
Members	1,356
Associate Members	540
Junior Members	357
Student Members	91
	2,372

Report of the Committee on Research

The Report of the Committee on Research for 1936, a Review and Forecast, was prepared by W. A. Danielson, Chairman, and F. C. Houghten, Director of the Research Laboratory, and was presented in abstract by Chairman Danielson.

The Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS held four meetings during 1936. The retiring committee for 1935 met in Chicago during the 42nd Annual Meeting of the Society and approved a budget prepared by the Finance Committee to be submitted for final approval by the 1936 Committee, and transacted other business pertaining to the previous year's activity.

An organization meeting of the 1936 Committee followed, at which time the new Chairman for 1936 was elected, the budget proposed by the retiring committee was adopted, and a research and finance program for the year was discussed. The committee also considered a number of matters of interest to the Laboratory, including the appointment of several Technical Advisory Committees. Two meetings of the Committee were held during the Semi-annual Meeting of the Society at Buck Hill Falls, Pa., when a number of matters pertaining to a reorganization of the Technical Advisory Committees and methods of financing the work of the Committee received extended discussion. The closing meeting of the 1936 Committee will be held in St. Louis during the 1937 Annual Meeting of the Society.

Financial Position Improved

For the third successive year following the depths of the depression, the Committee worked under a slightly enlarged budget and closed the year in an improved financial condition. Besides carrying on a number of studies at Pittsburgh, work was continued in eight universities. The better financial outlook of the Committee during the latter part of the year, and the receipt of a $4,000 appropriation from the Council of the Society made possible an expansion in the Laboratory budget and a renewal of the Committee's successful practice of helping to finance cooperative research in a number of Universities, after a lapse of several years in such aid during the depression. This renewed financial relationship with the cooperative institutions has encouraged

an expansion in research activity which should greatly increase the results obtained during the coming year.

By far, the most important development of the work of the Committee during the year was the re-organization of the Technical Advisory Committees in accordance with a comprehensive organization plan shown in Fig. 1. According to this plan, the interests of the Committee on Research have been divided into two major and several minor divisions. The first major division includes all studies and developments dealing with air conditioning in their relation to the comfort and health of man, air conditioning applications dealing with animals and plant life, and the conditioning treatment of by-products from the same.

Organic Division

This major branch, known as the Organic Division, as contrasted with a similar "inorganic" division, was organized under W. L. Fleisher, acting as assistant chairman of the Committee on Research. Under the sub-division of "Humans" are included any studies made by the Research Laboratory dealing with the desirable atmospheric conditions for human comfort and well-being, both in the summer and winter, the effects of drafts on comfort, the relation between odors and the volume of ventilation in occupied spaces, and the treatment of various diseases either for curative effect or for optimum comfort condition of the patient receiving other treatment.

Inorganic Division

The Inorganic Division was organized into two major sub-divisions covering "fundamentals" and "performance," headed respectively by Thomas Urdahl and E. W. Goodwin, acting as assistant chairmen of the Committee on Research. The Fundamentals Sub-division gives consideration to any fundamental knowledge pertaining to materials used in building construction or the manufacture of equipment. Such fundamental physical phenomena as heat, its transfer, and the physical and chemical qualities of the air are included in this sub-division of research activity. The Performance Sub-division deals with the performance or service of systems of heating, ventilating and air conditioning, and other equipment used in such systems, together with such practices as the generation of heat from fuels and methods of obtaining mechanical cooling.

For want of other location, two other activities of the Committee have been considered in parallel with the Organic and Inorganic Divisions, namely, the subjects of "climate in relation to the health and well-being of man," and "psychrometry."

Technical Advisory Committees

Under these major and minor divisions of the interests of the Committee on Research have been developed twenty-one Technical Advisory Committees, while several others have been proposed. With this comprehensive sub-division of the interests of the Committee, it is hoped that when any new question needing laboratory, library or committee investigation is proposed it may be immediately allocated to a Technical Advisory Committee already organized, thus avoiding delays in organizing such work, as well as the overlapping of responsibilities of Technical Advisory Committees appointed in haste when the need for the same arises.

A great deal of time and energy were required during the year in developing this re-organization and the appointment of the Technical Advisory committees. This naturally resulted in a delay in the organization of the committees themselves and of their several programs of operation, and hence, the reported activities of the committee in the appendix to this report will in some cases indicate little activity beyond organization. This has not, however, resulted in any delay in any of the projects assigned to the Research Laboratory or cooperating institutions by the Committee on Research, which have been carried on as energetically as finances and personnel have permitted. It is hoped that with the re-appointment of these same committees during the coming year their work will go forward without interruption and with improved performance.

Study of Finance Methods

The method of helping to finance the research work of the Committee by voluntary subscriptions from industry was given a great deal of consideration, and a

number of possible methods of organizing such efforts for the year were considered. With the improved economic condition of industry it has been thought that while the time for such a campaign has not yet been reached, the ground work should be laid for carrying on such a campaign in the very near future. With this in view, considerable effort has been made in the organization of the work of the Finance Committee, with a view of placing before prospective contributors a comprehensive program of work to be carried on and facts concerning the value of the work of the Committee on Research to this branch of the engineering profession and industry.

While the time was not considered ripe to push this effort to an ultimate conclusion during the year, a number of worthwhile results have accrued, including the receipt of $1,425 in earmarked contributions and $100 in non-earmarked contributions. It is believed, however, that these contributions should not be considered as a measure of the fruitfulness of the work carried on during the year, but that more comprehensive contributions should result during the early part of 1937 from the effort already expended.

Educational Publicity

Paralleling the development of the work of the Finance Committee, a Publicity Committee has been organized in order to properly place the importance and value of the Committee's work before the industry and financial institutions profiting by the growth in air conditioning. In this connection, the services of Brewster S. Beach, as publicity counsel, have been engaged, who has already carried on a number of activities which it is believed are approaching fruitfulness. While the results of these activities are not demonstrated in the financial returns to date, it is believed that the ground work has been laid, as a result of which further developments will take place during 1937.

Altogether, the work of the 1936 Committee may be considered as a transition in the life of the Research Laboratory, during which reorganization activities have been carried on aiming to return the Laboratory from a "depression" to a normal economic basis, and it is hoped that the success reflected in the results to be obtained during the next few months will amply demonstrate the wisdom of these efforts.

VITAL PROBLEMS IN RESEARCH FOR THE COMING YEAR—A FORECAST

Many vital problems face the Committee on Research for the coming year. Chief of these are the extension of knowledge as to how air conditioning can be utilized more effectively in the treatment of disease, the broadening of investigations and the development of new facilities and methods for testing insulation and further work in determining standards for summer comfort cooling embracing the whole subject of the physiological aspects of air conditioning, heating and ventilating.

Emphasis on Insulation

Considerable emphasis it is expected will be placed on the problems of insulation as it is felt these are fundamental to all branches of the industry. Among the subdivisions of this work may be mentioned studies of heat losses and gains through glass, correct practice for the use of glass brick in buildings, commercial development of water covering for roofs as insulation and to lengthen roofing material life, best ways of insulating glass in buildings to prevent infiltration of cold in heated buildings and further studies of glass as a "heat trap."

Special attention needs to be given to methods of installing insulation in buildings, particularly homes, to avoid settling, provide tight fitting and in other ways to eliminate or minimize the deleterious effect of humidified air from within the building itself on the insulation materials themselves.

These studies must encompass investigation of double glass windows and other insulation methods including reflective materials, board insulation and fill-type insulation. More advanced methods for testing all insulation materials as well as testing standards will be required in the immediate future. The importance of this work is especially apparent with the rapid growth in new building construction already approaching boom proportions. Serious monetary losses and damage to builders' and engineers' reputations may be the result of failure to improve scientific knowledge of the subject of insulation materials including methods for their correct application.

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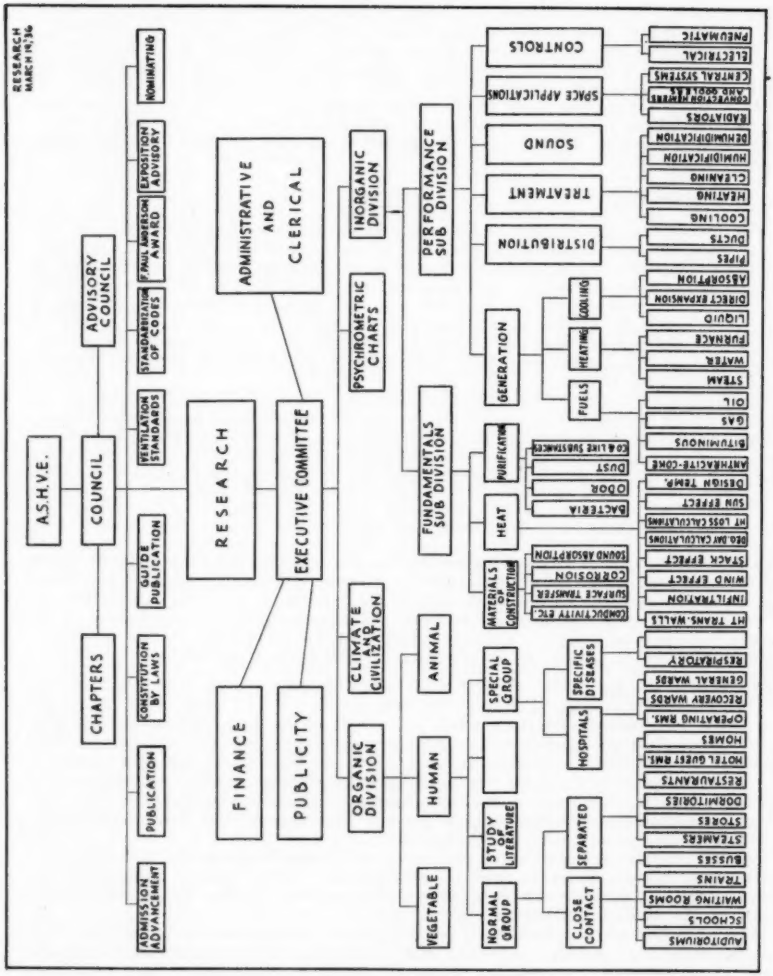

FIG. 1. FUNCTIONAL ORGANIZATION CHART.

Thirty Problems Listed

The Committee on Research has listed some 30 subjects that are desirable for study in all fields of heating, ventilating and air conditioning. Among those not already mentioned is the study of what constitutes a draft, recognizing that this subject is one which causes complaints from the public in ventilating and air conditioning systems. The physiological effect of the so-called draft and its relation to a feeling of comfort or discomfort will be investigated. This subject on the other hand, is also bound up in the broad problem of air distribution. The Committee expects to study further the proper method and place of introduction of air for room cooling and the safe limits of difference in temperature between the room and incoming air. Also the practical limitations of air velocity and difference in temperature between room temperature and air delivery temperature in the use of unit ventilators for heating and ventilating school rooms.

Related to these problems also is the question of noise generated in air distributing systems and the whole broad subject of noise in relation to heating and ventilating both in ducts and through air discharge outlets. Steam and hot water heating systems should come in for a special share of attention. Included in this work is research on the calculated drop in temperature through a radiator in forced circulation hot water heating, determination of the simplest and lowest-cost low pressure steam and hot water heating systems for small homes, and use of small pipes in forced hot water circulation systems.

The increasing complexity of air conditioning and the arts allied to it, demand a type of fundamental research that will help to stabilize the engineering and scientific side of these industries in the face of growing public demand for air conditioning applications of many new kinds. Research must keep pace with the rapid increase in the tempo of modern selling methods in order that heating, ventilating and air conditioning may march forward in orderly fashion with a secure foundation of basic research facts behind them.

PAPERS RESULTING FROM RESEARCH DURING 1936

Twelve technical papers, resulting from research carried on in Pittsburgh or in the cooperating institutions under the several Technical Advisory Committees were prepared for presentation at meetings of the Society and published in the A.S.H.V.E. JOURNAL. They are listed below and will be referred to by number in the discussion of research projects carried on by Technical Advisory Committees in the latter part of this report.

1. Thermal Properties of Concrete Construction, by F. B. Rowley, A. B. Algren and Clifford Carlson. A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 33.
2. Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet. A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 215.
3. Performance of Fin-Tube Units for Air Heating, Cooling and Dehumidifying, by G. L. Tuve. A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 99.
4. Ventilation Requirements, by C. P. Yaglou, E. C. Riley and D. L. Coggins. A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 133.
5. Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg. A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 263.
6. Study of Summer Cooling in the Research Residence Using Water from the City Water Mains, by A. P. Kratz, M. K. Fahnestock, S. Konzo and E. L. Broderick. A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 193.
7. A Field Study of the Heat Requirements of a College Building, by F. E. Giesecke and W. H. Badgett. A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 383.
8. Heating Requirements of an Office Building as Affected by Weather Conditions, by F. C. Houghten and Carl Gutberlet. A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 391.
9. Condensate and Air Removal Rates in a Vacuum Heating System, by D. W. Nelson. A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 241.
10. Thermal Properties of Concrete Construction, by F. B. Rowley, A. B. Algren and Robert Lander. A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 33.
11. Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, Cyril Tasker and Carl Gutberlet. A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 145.
12. Fever Therapy Induced by Air Conditioning, by F. C. Houghten, M. D. Ferderber and Carl Gutberlet. A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 131.

ACTIVITIES OF TECHNICAL ADVISORY COMMITTEES

During the year 21 Technical Advisory Committees were appointed to consider as many different sub-divisions of technical knowledge on which this branch of engineering profession is based. As stated, a number of these committees have been appointed only long enough to perfect an organization, and little in the nature of technical

results are available from their efforts. However, in a few cases work has been continued under committees appointed in former years and under which definite research has been carried on. The personnel of each of the Technical Advisory Committees working under the Committee on Research, and a discussion of the work considered, organized, or carried on, during the year, are given.

Technical Advisory Committee IF-1—SOUND IN RELATION TO HEATING AND VENTILATION—J. S. Parkinson, *Chairman;* C. M. Ashley, C. A. Booth, V. O. Knudsen, R. F. Norris, C. H. Randolph, J. P. Reis, F. R. Watson.

This committee has done very little work during the course of this past year. It feels that its principal function was fulfilled in collecting available information for the use of the Society and recommending the necessary research to complete the desired data. There is very little research activity along these lines, and the principal efforts of the committee have been directed toward interesting different organizations in such research.

The committee has prepared a table of noise levels which are considered typical for various room locations, with the range of variation which is likely to be encountered. On the basis of these tabulated levels, the committee has recommended certain maximum permissible figures for the noise of equipment in such spaces. The committee has cooperated with Mr. Hotchkiss and the GUIDE Publication Committee in revising the Chapter on "Sound" for the 1937 GUIDE. It has likewise developed a research program aimed to make possible the rating of the noise generated by ventilating and conditioning apparatus.

The general situation is very much the same as it was when the committee met in July at Buck Hill Falls. We are in a position to say what noise levels are permissible and are beginning to have data available as to the reduction encountered through the ventilating system with and without absorbent linings. Lacking data as to the original noise created by the equipment, it is at present impossible to specify the levels in the finished installation with any degree of accuracy. This is a problem which clearly calls for collective action on the part of the manufacturers and the Society.

Technical Advisory Committee IF-2—CORROSION IN HEATING SYSTEMS—J. H. Walker, *Chairman;* E. L. Chappell, W. H. Driscoll, C. A. Dunham, L. B. Miller, R. R. Seeber, C. M. Sterne.

The principal function of this committee has been to cooperate with the Michigan College of Mining and Technology in the study of corrosion of ferrous metals in heating systems. The project has been under way since June, 1933, and papers in the nature of progress reports were presented by Professor R. R. Seeber, who is in charge of the project, at the 1934 Meeting and at the 1936 Summer Meeting[6, a] of the Society.

The general purpose of the study is to determine the effect on corrosion of various amounts of carbon dioxide and oxygen in the steam supply. Due to the complex and difficult nature of the observations, it is expected that at least another year will elapse before the final results can be presented.

Up to midsummer of 1936, the work was entirely financed by the College. Recently a fund of nine hundred dollars for its partial support during 1936-37 was subscribed by several public utility companies engaged in district heating, but the major portion of the financial support is still being contributed by the College.

INSTITUTIONS COOPERATING WITH THE COMMITTEE ON RESEARCH

Agricultural & Mechanical College of Texas

Comfort Requirements in Summer Cooling.
Dissipation of heat from pipes buried in ground with reference to cooling of buildings.
Efficiency of water jet pumps as used in central forced circulation water heating systems.

Case School of Applied Science

Heat transfer of finned tubes with forced air circulation, including dehumidification data.

Harvard School of Public Health

Minimum outdoor air requirements for ventilation, particularly with relation to odors.

[a] Footnote references refer to numbers of papers listed on page 13.

University of Illinois

Direct and Indirect Radiation with Gravity Air Circulation.
Cooling of buildings in Summer.
Air Conditioning in the Treatment of Diseases.

Marquette University

Methods of air supply and distribution in air conditioning.

Michigan College of Mining & Technology

Corrosion in steam heating returns under controlled laboratory conditions, with relation in composition of the water, vapor and gas handled.

University of Minnesota

Heat Transmission through Building Materials.
Method of testing dust filters.

Ontario Research Foundation

Summer cooling comfort requirements.

Princeton University

Measurement of Air Flow through Exhaust Grilles.

University of Wisconsin

Air and Condensate Removal in steam heating returns.
Methods of air supply and distribution in air conditioning.

Technical Advisory Committee OH-3—AIR CONDITIONING IN THE TREATMENT OF DISEASE—Dr. E. V. Hill, *Chairman;* N. D. Adams, J. J. Aeberly, John Howatt, A. P. Kratz, V. L. Sherman, E. L. Stammer.

The report of this committee at the present time can be only one of progress, although it is possible that there will be some research data of interest by the time of the Annual Meeting in January.

Considerable unavoidable delay was experienced in getting the program at the Research and Educational Hospital of the Medical College of the University of Illinois under way. It was first necessary to build an experimental air conditioning system to be used in connection with an experimental hospital room. This was accomplished through the generous support of manufacturers in supplying the equipment and in many cases aiding in installing and putting the same into operation. Otherwise, the installation and remodeling was carried on wholly by the hospital staff. In spite of these difficulties, the installation was finally completed and the apparatus finally put into operation on the first of November. It has since been operating continuously and in a very satisfactory manner.

The set-up at the Research Hospital consists of an equipment room and a small two-bed ward. The two rooms are adjacent. The equipment room contains supply and exhaust fans, duct work for recirculating any or all of the air or taking any part required from out-of-doors, special filters for air cleaning, a 3-ton compressor with shell and tube cooler, circulating pump, and fin type coils in the air passage to the supply fan, a silica gel dehumidifier for reducing the moisture content of the air, a steam jet humidifier for increasing the humidity, all under automatic control.

The ward is approximately 12 by 14 ft in area with no outside windows; it is insulated with three inches of cork; the temperature range that can be maintained is from 50 to 100 F dry-bulb, 20 to 90 per cent relative humidity. The staff conducting the research work consists of Dr. R. W. Keeton, head of the Department of Medicine at the Research and Educational Hospital, Dr. F. K. Hicks, and Mr. N. Glickman, with some assistance from the chairman of the committee.

The preliminary program undertaken is the exploration of the physiological reactions of normal and abnormal cases under various psychrometric conditions. At the present time this program includes: (1) the determination of cardiac output, i.e., the number of liters of blood per minute that leaves the heart, which is accomplished by the injection of a harmless dye into the venus blood stream and timing its circuit and appearance in the arterial flow; (2) skin temperature observations are made at each test, at five points on the body, namely, the forehead, both forearms, and both thighs; (3) the metabolic rate is also determined, both the oxygen consumption and the CO_2 output being measured. The program will be limited to these observations for the present. Later it will be amplified.

In making the tests referred to, the research staff is following a very definite

psychrometric procedure, starting at a given wet-bulb temperature as near the saturation curve as possible; then, the tests are continued with increasing temperature and lowered humidities, staying always on the wet-bulb curve selected. When the physiological responses on this curve are completed, another will be selected, possibly three in all; then, the same procedure will be followed in exploring dew-point lines on the chart, also relative humidity lines, and finally, the dry-bulb lines. In this way it is hoped to find the physiological effect of different air conditions, and it should be possible by following this procedure to properly evaluate such efforts, whether they are due to dry-bulb temperature, to relative humidity, or to any other psychrometric function.

Technical Advisory Committee IP-4—ATMOSPHERIC DUST AND AIR CLEANING DEVICES—H. C. Murphy, *Chairman;* M. I. Dorfan, S. R. Lewis, G. W. Penney, D. C. Simpson, W. O. Vedder.

The work of this committee, formerly covering all phases of the dust and air pollution problem, was divided during 1936, this committee continuing to interest itself in air cleaning devices as applied to ventilation and air conditioning problems, while the Technical Advisory Committee on Atmospheric Impurities and Resulting Safety and Health Requirements is now working on other phases of the subject.

This committee has continued its work on the formation of a code for the selection and rating of air cleaning and dust eliminating devices intended to supply satisfactory air for dust hazardous occupations. This is intended to supplement the Code for Air Cleaning Devices Used in General Ventilation Work, which the Society has already adopted as a standard.

The committee worked with the United States Bureau of Standards in the development of new test procedures and had hoped that these developments would be sufficiently advanced to warrant a paper at the January meeting. It now seems advisable, however, to postpone this paper to a later date. The committee has endeavored to keep in touch with new developments in this field and has attended wherever possible meetings of scientific and engineering associations where these matters have been under discussion.

It is still felt that the committee's suggestion of a clearing house for information of this kind is worth considering. There are innumerable investigations and reports of great importance to our profession which become available every day but nowhere is this information tabulated and correlated so that it will have maximum value for our members and the public in general.

The investigations of Mr. O. C. Durham of Chicago as to distribution of pollen in the air has enabled engineers to provide pollen-free spaces and considerable relief of seasonal hay-fever, bronchial asthma and similar allergic disorders. The aerial survey of air-borne organisms conducted by the Lindberghs in 1933 for the U. S. Department of Agriculture has brought new conceptions as to the transmission of plant diseases. The recent report of the University of California on the "Distribution of Disease by Trans-Pacific Airplanes"; the recently published investigation of the Harvard Observatory covering particle sizes of dust in the outer atmosphere, and many other worth while investigations appear daily, without any attempt to correlate them.

The committee feels that an extension of the work done in the "Correlation of Thermal Research" could be attempted at the Research Laboratory in Pittsburgh without undue cost and with real benefit to the Society and the public in general.

Technical Advisory Committee OH-5—AIR CONDITIONS AND THEIR RELATION TO LIVING COMFORT—C. P. Yaglou, *Chairman;* J. J. Aeberly, R. R. Sayers, C.-E. A. Winslow.

The cooperative study between the Society and the Harvard School of Public Health on minimum outdoor air requirements for ventilation was completed this Autumn. A progress report [4] on this work was presented at the last Annual Meeting in Chicago, and the final report will be available in the near future.

Provisional values given in the preliminary report were reinforced with additional observations during the year. Among the new points studied are the influence of winter humidification and spray washing on odor removal, rate of disappearance of various odors in unventilated rooms, and the effects of temperature, humidity, seasonal changes, etc., on the intensity of body odor. The conventional spray humidi-

fier, using clean water at the rate of 15 gpm per 1000 cfm of air circulated, was found to be fairly efficient in odor removal, allowing a great reduction in the outdoor air supply, with a corresponding increase in the recirculation rate up to 85 per cent under the experimental conditions.

Owing to pressing work in other directions, the committee feels that the determination of ventilation requirements for smoking rooms and methods of removing tobacco smoke odors had better be deferred for some future time or set aside for solution by manufacturers of air conditioning equipment.

A concerted effort should be directed toward the solution of the problem of summer cooling for comfort, and of the factors contributing to the causation of drafts. Much valuable work on these two problems is now being done by the Society's Research Laboratory and other interested institutions. The program should, however, be modified or expanded to deal with physiological rather than physical aspects, i.e., the ability of individuals to meet sudden changes in the physical components of the air without disturbance in the physiological balance. By far the greatest effort should be directed to studying the nature of physiologic response to sudden and frequent changes in temperature and humidity, when the body is warm and perspiring, and the air conditions most conducive to comfort with the least adaptive response.

In this connection, the Chairman of the Committee is strongly in favor of a thorough study of adsorption dehumidification methods and of the possibility of securing comfort in hot weather largely by reduction of humidity with but a few degrees drop in temperature. Considerable evidence was obtained during the past five years to justify a thorough study along such lines.

The Committee wishes to call attention to two important publications during the year, that of T. Bedford on "The Warmth Factor in Comfort at Work," and those of Winslow and his associates at the John B. Pierce Laboratory of Hygiene. Both of these studies deal essentially with the assessment of thermal factors influencing human comfort.

Technical Advisory Committee IP-6—FRICTIONAL RESISTANCE TO FLOW OF AIR IN SMALL DUCTS AND FITTINGS—J. H. Van Alsburg, *Chairman;* C. A. Booth, S. H. Downs, R. D. Madison, L. B. Miller, L. E. Smith, C. M. Humphreys.

This committee was organized during 1935 to make a thorough study of the available data on frictional resistance in all types of ducts and fittings, particularly of small sizes. This might be considered a check on the friction chart now appearing in the A.S.H.V.E. GUIDE. The committee met during each convention following its appointment and outlined a program including plans for a laboratory set-up which has since been built up and used at the Research Laboratory in Pittsburgh.

The first series of tests consisted of determining the resistance in an 8-inch round pipe. This set-up consisted of 10-ft sections giving a total length of 64 ft of straight duct, followed by an elbow, a section of straight duct, another elbow, and another section of straight duct equipped with a throttling cone. This duct system was attached to a 12-inch section, which was attached to the blower. The 12-inch section was equipped with a suitable orifice plate and pitot tube installation for making volume flow readings, and the necessary air straightening devices. After the first tests were reported, giving results differing from other organizations, the data were checked to the Reynolds number and all bibliography was checked to determine the reason for the variation between the new data and that published in the friction chart. The values of the factors obtained from the Laboratory's test data are lower than any other data found on galvanized pipe of the same diameter. This difference may be due in part to the use of smoother duct by the Laboratory, but probably to a greater extent to large factors of safety used in the friction chart.

The committee has developed plans for continuing the work at the Research Laboratory. These plans include studying square and rectangular ducts of equivalent capacity to the 8-inch round duct, with various elbows and other fittings. A similar study will then be made on at least one larger size of duct with different dimensions.

Technical Advisory Committee IP-7—REFRIGERATION IN RELATION TO AIR TREATMENT—M. K. Fahnestock, *Chairman;* E. A. Brandt, John Everetts, Jr., E. D. Harrington, E. D. Milener, K. W. Miller, E. B. Newill, F. G. Sedgwick, J. H. Walker.

The studies in summer cooling carried on at the Research Residence by the University of Illinois in cooperation with the Research Laboratory and the *National Warm Air Heating and Air Conditioning Association* were continued through the

summer of 1936 for the purpose of completing the test program approved and started in 1935. This program was to determine the operating characteristics of the resident cooling plant when using water at three different temperatures; namely, 58, 52, and 45 F, in extended surface cooling coils. The summer of 1935 was relatively cool, and only the series of tests using water from the city supply mains at a temperature of 58 F was completed during that summer. A complete report [6] of that work was made before a joint meeting of the Society and the *National Warm Air Heating and Air Conditioning Association,* held during their annual meetings in Chicago in January.

The unusually warm summer of 1936 permitted the completion of the two remaining series of tests, using water with temperatures of 52 and 45 F. The arrangement of the test plant, with the exception of the refrigerating equipment for supplying chilled water, was the same as during 1935. The condensing unit and water cooler for supplying chilled water were located outside of the residence in order to prevent the introduction of heat from mechanical equipment which was not used during 1935, when the cooling water was taken directly from the city mains. Over the entire period of time requiring cooling the air circulating fan operated continuously and outdoor air equivalent to one air change per hour was brought into the house for ventilating purposes. The indoor temperature control was effected through a dry-bulb thermostat set at 80 F, which served to start and stop the circulation of chilled water through the cooling coil by the operation of a water circulator. The temperature of the chilled water entering the cooling coil was regulated by a thermostatically operated water mixing valve. The same amount of cooling coil surface, eight rows deep in the direction of air flow, was used during all tests in 1935 and 1936.

The committee has the following program under consideration for possible future study:

1. A study of evaporative cooling of residences, involving a test program in the Research Residence.
2. A study of the use of a forced-draft cooling tower, including the design of same, for the purpose of reducing the quantity of cooling water required with mechanical refrigeration. This might also include a study of the use of evaporative condensers.
3. Consideration for a survey of existing data and information available on residence cooling installations in an attempt to determine why there is such a large variation in the recommended and actual capacities of refrigeration equipment for moderate size houses.
4. A continuation of one season of mechanical cooling, if and when the Research Residence is insulated.

Technical Advisory Committee IF-8—HEATING AND COOLING REQUIREMENTS OF BUILDINGS—O. W. Armspach, *Chairman;* P. D. Close, W. H. Driscoll, H. M. Hart, P. E. Holcombe, V. H. Hunter, H. H. Mather, E. C. Rack, F. B. Rowley, R. J. J. Tennant, J. H. Walker.

This committee has been most active over a period of years in making a comprehensive study of the heating requirements of buildings. In this connection a large number of studies were initiated and carried on at the Research Laboratory and at several cooperating universities. During the past year the estimation of cooling requirements of buildings has been added to the responsibilities of the committee.

There has been little change in the active program of the committee during the current year, but a number of the activities outlined for a long time program have been continued. Additional data collected in the Grant Building were analyzed by the Research Laboratory and a paper [8] was presented to the Semi-Annual Meeting of the Society. Additional data remain to be analyzed in connection with this study which will result in another report at some future date.

A study of the heating requirements of Sibley Dome, one of the Cornell University buildings, has been continued. Data in connection with this study were collected at an earlier date by Mr. Ward, Superintendent of Buildings and Grounds at Cornell University, and the results are now being analyzed by a sub-committee with a view of presenting a paper to the Society.

A study was made of the heat requirements of one of the buildings at the Agricultural and Mechanical College of Texas by Professor Giesecke. This work extended over the past heating season and resulted in a paper [7] presented at the Semi-Annual meeting of the Society. Professor Giesecke is continuing this study.

The study of heat transfer through various types of concrete construction was continued at the University of Minnesota by Professor Rowley in cooperation with

the Society and the *Portland Cement Association.* A paper [1] resulting from this study was presented at the last Annual Meeting of the Society, and a second paper [10] will be presented at the coming Annual Meeting.

A study of condensation and air return rates in the heating of a large school building was continued at the University of Wisconsin in cooperation with the Laboratory, resulting in a paper which was presented at the Semi-Annual Meeting of the Society.

The study of the use of the "degree-day" method of estimating fuel consumption was continued by a sub-committee. A report on this phase of the work is expected at an early date.

The various activities of the committee will be continued through 1937. In addition to these continued activities, a further study of heat transfer through roofs and walls of buildings as affected by heat capacity and solar radiation under practical conditions has been approved by the Committee on Research, and it is hoped that finances will allow an active investigation of the subject during the coming summer months.

Technical Advisory Committee IP-9—DIRECT AND INDIRECT RADIATION WITH GRAVITY AIR CIRCULATION—M. K. Fahnestock, *Chairman;* H. F. Hutzel, J. P. Magos, J. W. McElgin, J. F. McIntire, D. W. Nelson, R. N. Trane, T. A. Novotney.

Although the work of the committee has been considerably curtailed during the past year, some active research was continued at the University of Illinois.

The work to determine the effect of the residual energy stored in different types of radiators and convectors upon the air temperature in a room when the steam supply is turned off has been completed and the results are being compiled in a paper for presentation to the Society. This work was done in the room heating testing plant under conditions approximating those found in actual service. The tests were made with two types of direct cast-iron radiators, two convectors with two types of cast-iron heating units, and two convectors with two types of non-ferrous heating units. The results indicate that there is an appreciable difference in the rate at which the room cools with the different units.

The remodeling of the room heating testing plant for the purpose of providing one large test room 15 ft by 18 ft instead of the two small test rooms 9 ft by 11 ft has been started, and the work is approximately 25 per cent completed. This work will be carried to completion as rapidly as possible.

The work in the warm wall test booth providing for a study of the inlet and outlets of convector heater cabinets with reference to their effect upon the heat output or capacity of the units has been at a standstill for the past year. It is hoped that material progress can be made on this investigation during the next few months and that possibly the project can be completed and the results presented at the next Annual Meeting of the Society.

Technical Advisory Committee IP-10—HEAT TRANSFER OF FINNED TUBES WITH FORCED AIR CIRCULATION—F. B. Rowley, *Chairman;* H. F. Hutzel, C. E. Lewis, E. J. Lindseth, R. H. Norris, W. E. Stark, G. L. Tuve.

At its meeting in Chicago in January, the committee approved further plans for research to be carried on at Case School of Applied Science in cooperation with the Research Laboratory during 1936. In accordance with this program, the work has been directed toward the study of dehumidification by cooling surfaces.

To make this study possible, the wind tunnel test equipment at Case School of Applied Science, where the work has been in progress for five years, was therefore further improved to make it possible to produce any wet-bulb-dry-bulb combination ordinarily encountered in comfort cooling, and to accurately control and maintain any desired condition.

A report giving the results of the hundred or more dehumidifying tests made at Case during the year is now being completed and will be published soon. This report indicates that while the work which has been done on this subject to date, including both the results of other investigators and the two papers [3] on this subject previously presented to the Society, is convincing with regard to certain aspects of the problem, there is much yet to be done. Several methods of rating and of calculation have been developed or suggested, but it is important that these be correlated, and that the fundamental laws of surface dehumidification be codified and made a part of the recognized technical equipment of the air conditioning engineer.

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The research project sponsored by this committee is therefore proceeding along two lines: (1) experimental determination of the relative accuracy of various methods already in use for expressing the performance of dehumidifying coils; and (2), more accurate analysis of variables, the effects of which are very imperfectly known at present.

The work of the committee has been greatly assisted by active cooperation of the Research Laboratory and of manufacturers and engineers in the industry. Personal contacts have been made with most of the laboratories which have been actively studying this problem in the last two years. A special effort is being made to correlate the present experimental project with the work which has been done by other investigators, mostly outside the Society.

It is hoped that a uniform and simple basis for rating fin-coils and for computing surface heat-transfer may finally be evolved and made a part of the Codes and recommended procedures sponsored by the Society.

Technical Advisory Committee X-11—PSYCHROMETRY—F. R. Bichowsky, *Chairman;* C. A. Bulkeley, J. A. Goff, E. V. Hill, F. G. Keyes, A. P. Kratz, W. M. Sawdon.

This committee was appointed during the year to review the psychrometric data now in use and recommend any desirable changes with a view of adopting standard values for general use throughout the industry. The committee has given some consideration to the problem and has organized its work into the following sub-divisions:

(1) *a.* Recommendation of definitions of the various psychromatic quantities including the definition of the standard temperature scale, probably the International scale.

b. The definition of the temperature of the gas which is a matter of some practical importance, since there seems to be a rather common failure to recognize that the thermometer in a gas stream may measure, not the temperature of the gas, but some sort of a mean between the temperature of the gas and the walls.

c. The definition of humidity both relative and absolute. It seems to be most usually forgotten that the usual definition of relative humidity in terms of relative partial volume or partial pressure of water vapor in air has no precise meaning except for a mixture of perfect gases.

d. The definition of dew, total heat, enthalpy, wet-bulb temperature, temperature of adiabatic evaporation, etc.

(2) Pick out standard Psychrometric data. This committee was established after it was determined that the International Committee on Steam Tables had about completed its work. The feeling was that these new Tables would for the first time give an authoritative source of data on the pressure, enthalpy and volume of water vapor. There is comparatively little accurate direct experimental data on the study of air-stream mixtures and it may be that one of the problems of the committee will be to recommend a program for experimental research to fill up this gap. However, it is believed that the increasing knowledge of the equation of the study of mixtures will allow the accurate calculation of psychrometric data for the air-stream mixture on the basis of data on pure steam and pure air. Dr. Keyes, a member of the International Committee on Steam Tables, has been asked to make a preliminary calculation on the source of data and on the method of reduction of data.

(3) Calculate very accurately a few points on the Psychrometric Table to compare with existing tables so as to discover if there are wide discrepancies as have sometimes been claimed. If this preliminary calculation shows that a complete recalculation of the Psychrometric Tables is necessary, the committee will require the services of a skilled calculator since such a complete recalculation is a very large job.

(4) The last function of the committee is to decide on the form of publication of its results, whether in the form of a psychrometric table or whether in the form of a psychrometric chart or charts.

Technical Advisory Committee OH-12—RAILROAD AIR CONDITIONING—A. I. Brown, *Chairman;* F. E. Giesecke, C. S. Leopold, S. R. Lewis, L. W. Wallace.

In recognition of the important place that railroad air conditioning has assumed, this committee was formed this year for the purpose of cooperating with the railroads in any technical studies of railroad air conditioning.

Air conditioning of railroad passenger cars has met with tremendous public approval. There can be no doubt that air conditioning has played an important part in the phenomenal increase in travel by railroad that has occurred during the past year. The experience of traveling in comfort in an air conditioned passenger car even in the hottest of weather has been a revelation to many people and has awakened in their minds the tremendous possibilities of the application of air conditioning for comfort in many fields beyond that of public conveyances. The investment of the railroads of the United States and Canada in air conditioning equipment has now amounted to 45 million dollars. The number of passenger cars that have been air conditioned now totals over 8,000.

Early in the year the Association of American Railroads set up plans for a comprehensive study of air conditioning of passenger cars, under the direction of Mr. L. W. Wallace, Director of the Division of Equipment Research of that Association. This study has proceeded along the following lines:

1. A large amount of factual data has been collected relating to the types and extent of air conditioning equipment being used by the various railroads, the capital investment and maintenance and operating costs.
2. A laboratory for testing air-conditioning units was set-up in the Mt. Clare Shops of the Baltimore and Ohio Railroad at Baltimore, Maryland, where tests have been conducted on the various types and makes of air conditioning units used by the railroads. These tests were made in accordance with the Standard Method of Testing and Rating Air Conditioning Equipment proposed by the Joint Committee of the A.S.H.V.E. and the *A.S.R.E.*, and other technical societies.
3. Dynamometer tests of the various driving mechanisms used in the operation of railroad air conditioning equipment have been conducted in the Mechanical Engineering Laboratory of the Ohio State University for the purpose of determining the mechanical efficiency of these mechanisms.
4. Hot room tests of fourteen air conditioned cars have been conducted in the Chicago Plant of the Pullman Standard Car Company. These tests dealt with the determination of the heat insulation value of cars of different type and construction and with the distribution of air and temperatures within the cars, and further served to coordinate measurements of cooling effect in the cars with the results of laboratory and road tests of the air conditioning equipment.
5. A five-day conference of railroad air conditioning engineers was held at Ohio State University early in the summer preparatory to participation in the investigation of air conditions being maintained in passenger cars in service. This conference was attended by engineers from 31 railroads of the United States and Canada on which the road tests were to be conducted.
6. In the road tests the air conditions in nearly 600 passenger cars were measured during 240,000 miles of travel. In response to requests for the opinions of passengers concerning the degree of comfort at the time when the tests were being made, more than 5,000 passengers submitted their comments.

It is hoped that this extensive research will make a valuable contribution to the progress in air conditioning as well as result in added benefit to the traveling public.

Technical Advisory Committee IF-14—CORROSION IN AIR CONDITIONING EQUIPMENT—A. E. Stacey, *Chairman;* M. L. Diver, M. S. Kice, F. L. LaQue, C. E. Lewis, R. M. Palmer, F. N. Speller, C. M. Sterne, R. T. Thornton, H. H. Young.

The committee was appointed a little over a year ago to make a study of the prevalence of a corrosion problem in connection with air conditioning equipment. The committee organized, held several meetings, and sent out questionnaires to users of air conditioning equipment to determine the extent to which corrosion was met with and the character of the problems involved.

These questionnaires resulted in definitely proving that there was not a corrosion problem involved in the use of spray type dehumidifiers and coil surfaces, except in certain localities. Water treatment, with service, seems to have satisfactorily solved this problem even in these districts where trouble is otherwise found.

Technical Advisory Committee on TREATMENT OF AIR WITH ELECTRICITY—C.-E. A. Winslow, *Chairman;* Vannevar Bush, W. H. Carrier, L. W. Chubb, W. D. Fleming, R. F. James, L. R. Koller, C. A. Mills, E. B. Phelps, C. R. Wait, W. T. Wells.

This is a new committee appointed during the year to take up the study of a most interesting phase of air conditioning. So far, the committee has only had an opportunity to organize, and hold a meeting January 13 in New York.

Technical Advisory Committee IF-18—AIR CONDITIONING REQUIREMENTS OF GLASS—M. L. Carr, *Chairman;* F. L. Bishop, A. N. Finn, E. H. Hobbie, R. J. Lillibridge, F. W. Parkinson, W. C. Randall, C. Tasker, R. A. Miller, G. B. Watkins.

This committee was appointed during the year and charged with a study of all problems involving the window and its setting in a building. A meeting of the committee was held in Pittsburgh on September 25, and gave consideration to the following program of research, and approved Item (1) for immediate study:

(1) Heat transfer through glass from air to air, including single, double, and triple glass, and the effect of spacing.
(2) Heat transfer through windows as affected by sash or other accessories of application.
(3) Heat transfer through different types of glass.
(4) Infiltration through windows for all kinds of applications, including weather-stripping.
(5) Solar radiation effects through windows and reduction in this effect by sun arresting appurtenances.
(6) Heat transfer through glass brick walls.
(7) Solar radiation through glass brick walls.

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The committee further asked the Research Laboratory to make a survey of possible data available on any phases of the subject as a result of work carried on in the laboratories of manufacturers or elsewhere. In making this survey, the following item should be emphasized:

(1) Find out what unpublished data, if any, on heat loss or transfer through glass, single, double and triple glazing, can be made available to the Research Committee. In this connection, the various glass manufacturing companies and the United States Bureau of Standards should be consulted.

(2) Find out what is being done in various laboratories on the subject that might be made available, in whole or in part, to the Research Committee. Advise with such laboratories as may be willing to cooperate as to scope and methods to be followed, with a view to obtaining data that the Society would be willing to publish in THE GUIDE.

(3) Find out where the A.S.H.V.E. Laboratory can carry on research or have research carried on, under its supervision, to obtain such data as may be needed for THE GUIDE on the subject of air to air heat transmission through single, double and possibly triple glazed windows.

Technical Advisory Committee IP-20—INTERMITTENT HEATING OF CHURCHES, AUDITORIUMS AND SIMILAR BUILDINGS—E. K. Campbell, *Chairman;* W. L. Cassell, E. F. Dawson, N. W. Downs, J. M. Robertson, J. H. Kitchen, A. H. Sluss.

This committee was appointed during the year and charged with a study of the time rate of heating and cooling buildings, together with the extra heat load during the heating up period, in relation to the structure or type of building. The committee has given very serious consideration to a number of directions of attack, including a study of the heat capacity or specific heat of materials commonly used in building structures, and a study of the problem by assuming a given temperature throughout the structure at the beginning of the heating up period, and the surface area and film conductance coefficients for the various walls, partitions, ceilings, and floors adjacent to the heated space.

Technical Advisory Committee IP-21—EFFECT OF ENTERING TEMPERATURE AND VELOCITY ON TEMPERATURE AND DISTRIBUTION OF AIR WITHIN AN ENCLOSURE—Ernest Szekely, *Chairman;* S. H. Downs, M. K. Fahnestock, D. W. Nelson, C. H. Randolph, J. E. Schoen, G. L. Tuve, J. H. Van Alsburg.

This committee began the year by continuing to lend its support to the cooperative study carried on under Dean Kartak at Marquette University in cooperation with the Research Laboratory. The lack of personnel other than undergraduate students made it seem futile to obtain worthwhile results on so complicated a problem.

As a result of continued pressure for results in this field, the committee expanded its program in November to include more intensive studies of the subject at the laboratory of the Barber-Colman Company at Rockford, Illinois, under the supervision of Professor G. L. Larson's Department of the University of Wisconsin, at Professor Larson's Department at the University, and by field observations.

The study at the Barber-Colman laboratory will at first be confined to fundamental observations of an air stream issuing from a plain opening of various sizes, shapes and locations, and with different air stream temperatures and velocities. This will be followed by similar studies with Multiblade type of grilles.

A study will also be carried on at the University of Wisconsin in connection with the system in the new Auditorium. These tests are to simulate those at the Barber-Colman laboratory within the limitations of the equipment, and if feasible will be extended to furnish comparison between the grilles in the side walls and discharge through a perforated ceiling.

The field observations will be made wherever possible. At the present time, the Chairman of the committee and Mr. Randolph have agreed to make certain observations in Milwaukee.

Technical Advisory Committee OH-22—COMFORT REQUIREMENTS FOR SUMMER COOLING—W. L. Fleisher, *Chairman;* A. E. Beals, F. R. Bichowsky, Thomas Chester, F. E. Giesecke, Elliott Harrington, E. V. Hill, R. E. Keyes, C. Tasker, C. P. Yaglou.

This committee was organized during 1935 to study the cooling requirements for summer comfort air conditioning. A paper[2] was presented to the Society at its Annual Meeting in Chicago, giving the results of a brief study made at the Research Laboratory during the summer of 1935. During 1936 the Committee continued the study at the Laboratory at Pittsburgh and organized additional studies to be carried on by Mr. Tasker of the Ontario Research Foundation at Toronto, Canada, and by

Professor Giesecke of the Agricultural and Mechanical College of Texas at College Station, Texas.

The study during 1936 was aimed to determine:

(1) The variation in the cooling requirements for summer air conditioning with different normal summer temperatures in different geographical regions in United States and Canada.
(2) To further indicate the range of temperature, humidity and air motion, or Effective Temperature, allowable for summer cooling.
(3) The allowable variation in relative humidity at a given Effective Temperature.
(4) The variation in the cold shock upon entering a cooled space, and the reaction upon entering the hot outside based upon the Effective Temperature maintained in the conditionad space, and variations in these effects with variation in relative humidity at the same Effective Temperature.
(5) The skin temperature changes upon entering the cooled space from the hot outside, and upon re-entering the hot outside for different Effective Temperatures indoors, and for different relative humidities at the same Effective Temperature.
(6) Allowable air velocities in summer cooling, and particularly, allowable velocities with high moisture content in the air, together with reasons for limiting high velocities; as an example, should such high velocities be limited because of excessive cooling on definite portions of the body, or due to the annoying effect of high velocities in disturbing papers or other materials on which a person may be working; or other more definite sensations of velocity experienced by a person?
(7) The sensation of draft and accompanying lowering of the skin temperature of various parts of the human body due to air velocities or drafts of varying velocities and air stream temperatures at various room temperatures; or in other words, a determination of what constitutes a draft, and the discomfort experienced as a result of the same.

During the summer of 1936 data [11] were collected showing a measurable, although not extremely large, variation in the required temperature for indoor summer cooling as between Pittsburgh and Toronto, Ontario, but no similar change in the requirement as between Pittsburgh and Texas. The three studies indicated that the indoor cooling requirements were based upon Effective Temperatures, and that little variation in comfort was experienced for rather wide variations in relative humidity at a desired Effective Temperature. While some observation was made of the cold shock upon entering a cooled space in relation to the conditions maintained indoors, this phase of the subject requires further study. Items 5 to 7 received little attention during the past summer, and are being continued by the committee, while Item 7, or a study of what constitutes a draft, is being continued throughout both the heating and cooling season of 1937.

The committee has given a great deal of consideration to the general subject of the relation of health and comfort of persons to their atmospheric environment, including both normal and abnormal or diseased persons. The committee has particularly given a great deal of consideration in interesting the medical profession in the use of air conditioning in the treatment of disease. In this connection, the committee was particularly able to cooperate with a number of medical centers, including cooperation between the Laboratory in Pittsburgh and St. Francis Hospital, where the Laboratory, through its past experience in studying the relation between physical reactions of persons to high temperatures, was able to successfully aid in the development of an air-conditioned "fever therapy box," now being used at the St. Francis Hospital. A paper [12] on the development of this box is being presented at the Annual Meeting in St. Louis.

Technical Advisory Committee IF-23—INSULATION—L. A. Harding, *Chairman;* E. A. Allcut, J. D. Edwards, E. C. Lloyd, R. T. Miller, E. R. Queer, T. S. Rogers, F. B. Rowley, W. S. Steele, C. Tasker, B. Townshend, G. B. Wilkes.

This committee was appointed during the year and charged with the making of a survey of the entire subject involving the testing and reporting of heat transfer values for, and the economic usage, of insulation. After organization, eleven members of the committee met with the Chairman of the Committee on Research and the Director of the Research Laboratory in Buffalo, N. Y., for an all day session.

The committee first agreed that the Society's present code for testing building insulation material for heat transfer needed revision to bring it up to date, and a sub-committee consisting of Professors Rowley, Allcut, Wilkes, Queer, and Mr. Houghten was appointed and asked to report back to the main committee.

The committee further gave consideration to the following questions concerning the usage of insulation, but deferred until a later date in making recommendations concerning the need for study of any of the particular subjects mentioned or concerning their order of importance.

(1) The effect of "regain moisture" on the conductivity of insulating materials.
(2) The effect of condensed moisture on the conductivity of insulating materials in a structure.

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(3) Effect of the thickness of an insulating material accepted for test on the conductivity value obtained.
(4) Effect of condensation on the surface of reflective insulation.
(5) Effect of fibre size and fibre arrangement within an insulating board on its conductivity.
(6) To what extent do radiation, convection and conduction play a part in the transfer of heat through an insulating material.
(7) Relation between the heat capacity of an insulating material on the rate of heat penetration through a structure in which it is installed.
(8) The degree of settling found for different "fill" materials after it is placed in the structure.
(9) The relation between the conductivity of "fill" insulating material and the density of the pack—What is the optimum density and lowest conductivity?
(10) Effect of sealing cracks around windows and doors in order to keep infiltering air out of spaces containing insulating materials on the heat transfer through the structure.
(11) Relation between the physical characteristics of the material as regards the capillary properties of the fibres in the structure and its behavior in use.
(12) A further study of the film conductance coefficients for air spaces, particularly where reflective surface insulation is used. Such studies should be made with a view of separating heat transfer to or from a surface by radiation and convection.
(13) Film conductance and air space coefficients should be given for both horizontal and vertical surfaces.
(14) Effect of radiation from hot metallic or other types of surfaces in a hot-plate testing apparatus on the conductivity value obtained for different types of materials. This may account for apparent discrepancies in conductivity values obtained for different materials by different investigators using different apparatus.
(15) Study of heat transfer through walls containing insulating materials in small electrically heated test houses.
(16) More emissivity values should be determined for different materials and tabulated in THE GUIDE.
(17) The corrosion action of insulation materials in the presence of moisture on other materials, including metals used in the structure.
(18) The adaptability and performance characteristics of insulation when applied to the modernization of older buildings.
(19) A study of the characteristics of insulation after long service in a structure.
(20) More information on the effect of applying insulating material of a given thickness having given conductivity characteristics to a structure.
(21) Effect of expansion or contraction and other physical changes in a building on the performance of insulating materials used in connection therewith.

Technical Advisory Committee X-24—CLIMATE AND ITS RELATION TO AIR CONDITIONING FUNDAMENTALS—Dr. C. A. Mills, *Chairman;* G. C. Dunham, James Govan, Dr. W. J. McConnell, C. F. Neergaard, F. M. Pottenger, Jr., E. L. Weber, C.-E. A. Winslow.

This committee, appointed during the year, is engaged in organization and the development of a program. An attempt is being made by mail to gather opinions from the members as to the major problems to be faced.

It is well agreed that much basic data yet remain to be collected on the physiologic response to physical environment before air-conditioning can be put on a firm and lasting footing. Some feel that quite radical alterations must be made in conditioning methods, and even principles, before wholly satisfactory results can be obtained. Closer investigation should be made of the use of radiant heat for comfort conditioning for both winter and summer. The physiologic effects of storm fluctuations in barometric pressure must be investigated, for there are strong hints that pressure control may before long be deemed of importance second only to control of temperature.

It seems imperative that the Society, through its Committee on Research, make every effort to speed up studies dealing with the fundamentals of air conditioning in the immediate future, so as to be better prepared for the period of marked expansion that lies in the years just ahead. Knowledge of the response of normal individuals to environment seems more important in this respect than does that obtained from the sick, since the major expansion of air conditioning will be in buildings occupied by normal individuals.

Technical Advisory Committee IF-26—ATMOSPHERIC IMPURITIES AND RESULTING SAFETY AND HEALTH REQUIREMENTS—Theodore Hatch, *Chairman;* J. J. Bloomfield, C. A. Booth, Philip Drinker, Dr. Leonard Greenburg, H. B. Meller.

So far, the work of this committee has been limited to a survey of the problem and the study of the different organizations interested in it. A meeting of the committee is being arranged for the near future.

During the past year there has been a great increase in the number of states offering technical services in industrial hygiene and at the present time 19 such bureaus are in existence. At least one engineer is attached to each office. There

are also in existence several important active committees engaged in reviewing and studying the problem. Among these may be mentioned:

1. Safety Code Committee on Exhaust Systems, *American Standards Association.* (Several sub-committees have been formed to prepare codes for different processes, one to prepare report on Fundamentals relating to Exhaust Systems.)
2. Advisory Committee on Toxicity of Dusts and Gases, *American Standards Association.*
3. Preventive Engineering Committee, Air Hygiene Foundation of America, Inc. (The preliminary report of this committee recommends certain research problems to the Directors of the Air Hygiene Foundation.)
4. Committee on the Prevention of Silicosis through Engineering Control. National Conference on Silicosis and Similar Dust Diseases. (This committee has several sub-committees engaged in preparing reports on the many different phases of the problem. The main report will be available soon.)
5. Committee on Ventilation and Atmospheric Pollution, *American Public Health Association* (Part 11, including several sub-committees, has to do with standard methods for the examination of the air; determination of dust, bacteria, gases and thermal factors affecting human comfort. Reports from this committee are already in print).

These activities indicate a growing realization of the need for the development of fundamental engineering methods of design to care for greater numbers of hazardous industrial processes. With respect to the functions of the present committee, it is essential that the problems in the field be carefully reviewed, that the work performed by existing committees be studied, and that the opportunities for cooperation with these committees be investigated, in order to make certain that the program adopted will yield the most useful results and not cause duplication of effort.

The general objectives of the committee include:

1. Collection and evaluation of existing published data pertaining to the industrial dust control problem.
2. Outlining and guiding fundamental research problems by the Society.
3. Fostering field studies for the collection of basic engineering data.
4. Action to bring about the preparation and publication of more technical papers in this field. A great deal of valuable engineering work is being done in industrial establishments, but very little of this is described in technical journals in a form that is helpful to others.

RESEARCH LABORATORY

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

STATEMENT OF INCOME AND EXPENSES

(Year Ending December 31, 1936)

INCOME

		Budget 1936	*Actual 1936*
1.	Balance from 1935 Committee	$8,178.62	$8,178.62
2.	Dues—Current and Prior Years	11,500.00	.....
3.	Guide	1,500.00	.....
4.	Heating and Ventilating Exposition	2,500.00	6,828.85
5.	Interest on Bank Balances	30.00	31.46
6.	Interest on Savings Account	70.00	63.76
7.	Interest on Society Endowment	400.00	600.00
8.	Contributions—Earmarked	.....	1,425.00
9.	Contributions—General	.....	100.00
10.	Special Council Appropriation	.....	4,000.00

EXPENDITURES

		Budget 1936	*Actual 1936*
1.	Travel	$ 1,200.00	$ 1,370.65
2.	Salaries—Laboratory Staff	9,500.00	8,760.87
3.	Salaries—Part-time Student Help	1,500.00	1,491.30
4.	Cooperative Research—General	2,675.00	1,012.50
5.	Cooperative Research—Earmarked	1,425.00	1,193.75
6.	Cooperative Research—Med. School	1,800.00	1,800.00
7.	Office Supplies and Expense	400.00	585.86
8.	Laboratory Supplies and Equipment	700.00	476.61
9.	Exhibits, Meetings, etc.	400.00	374.13
10.	Printing and Promotion	400.00	40.00
11.	Correlating Thermal Research	600.00	600.00
12.	Salary—Office of Chairman	900.00	882.82
13.	Promotion—Financial Campaign	2,500.00	701.10
14.	Contingency Fund	500.00	.....
	TOTALS	$24,500.00	$19,289.59
	Unappropriated Balance	3,678.62	
		$28,178.62	

Research Contributors—1936

Funds and Equipment

Aerofin Corp.
American Air Filter Co.
Barber-Colman Co.
Bryant Heater Co.
Canada Vulcanizer & Equipment Co.
Canadian General Electric Co., Ltd.
Carrier Corp.
Chamberlin Metal Weatherstrip Co.
Chicago Master Steamfitters Assn.
Clarage Fan Co.
Detroit Edison Co.
C. A. Dunham Co., Ltd.
Edison Electric Illuminating Co. of Boston
Fairbanks-Morse Co.
Julien P. Friez & Sons, Inc.
Frigidaire Corp.
General Electric Co.
Gilbert & Barker Mfg. Co.
Heating, Piping & Air Cond. Contractors National Assn.
Hoffman Specialty Co.
Johnson Service Co.
Johns-Manville Co.
Kelvinator Corp.
Kroeschell Engrg. Corp.
P. H. MaGirl Foundry & Fur. Wks.
McDonnell & Miller
Meyer Furnace Co.
Minneapolis-Honeywell Regulator Co.
Minneapolis-Honeywell Regulator Co., Ltd.
Mueller Brass Co.
National Warm Air Heating & Air Conditioning Assn.
Nash Engineering Co.
J. J. Nesbitt, Inc.
New York Steam Corp.
Philadelphia Electric Co.
Portland Cement Assn.
Rochester Gas & Electric Corp.
Rome-Turney Radiator Co.
Stannard Power Equipment Co.
Trane Co.
Tuttle & Bailey Co.
Union Electric Light & Power Co. of St. Louis
Ventilating & Air Conditioning Contractors Assn. of Chicago.
Westinghouse Electric & Mfg. Co.
Westinghouse Electric Co., Ltd.
L. J. Wing Mfg. Co.
Young Radiator Co.
York Ice Machinery Corp.

J. Lawrence DeNeille, St. Louis, Mo., former president of the *Heating, Piping and Air Conditioning Contractors National Association*, gave a brief word of greeting.

The secretary of the Society, A. V. Hutchinson, New York, presented the Report of the Council and the Report of the Secretary.

Report of Council

Since the 42nd Annual Meeting, the Council has held four meetings, one in Chicago, two at Buck Hill Falls, Pa., and one in St. Louis, Mo.

At the meeting on January 30, 1936, the Council organized and appointed its Committees, approved a new JOURNAL contract, selected St. Louis as the place for the 43rd Annual Meeting, approved depositories for Society funds, adopted the Budget for 1936 and appointed a new Certified Public Accountant; proposed a Sub-Committee on Air Conditioning Standards and authorized a survey on the relationship of air conditioning and public health.

At the June meeting, it was arranged to authorize payment of railroad expense for one executive officer selected by each Chapter as an official delegate to the Annual Meeting in St. Louis and arrangements were made for appointment of a Joint Committee by the Society and the *A.S.R.E.* to participate in the formation of a Code for Minimum Requirements of Comfort Air Conditioning.

The invitation of the Massachusetts Chapter to act as hosts for the Semi-Annual Meeting 1937 was accepted and it was decided to have the Society's 44th Annual Meeting during the fourth week of January, 1938, in New York, when the 5th International Heating and Ventilation Exposition will be held.

The Council authorized new advertising rates for THE GUIDE effective July 1, 1936, and appointed Ralph F. Duysters to the staff in accordance with the request of the Guide Publication Committee.

The Council made available a fund of $4000 to the Committee on Research for an enlarged program and appropriated $3000 for a cooperative program with the

New York Academy of Medicine provided a similar amount could be obtained from other sources to study the relationship of air conditioning and health. The Council nominated five members to serve on the Committee on Research for a term of three years.

The Council received a report from the special committee studying the relationship of the A.S.H.V.E. and *A.S.R.E.* and recommended that this Committee continue its work. The request of several Chapters that assistance be given to local organizations by the Society was considered and referred to the incoming Council.

The Council has carried on the business activities of the Society through the Officers and Committees and is glad to report that membership is at a peak figure through the addition of 490 new members this past year; that the finances are in excellent condition and that Charters have been granted for new Chapters in Montreal, Oklahoma City, and Texas.

Routine action was taken in connection with the resignations offered by members and cancellations of membership for non-payment of dues as required by Article B-II, Section 2. Life membership has just been granted to four members as follows: F. C. Black, R. L. Gifford, J. G. Hayes, F. W. Powers.

The Society has enjoyed a most successful year and plans are under way for enlarged activities for 1937.

Respectfully submitted,
THE COUNCIL.

Report of the Secretary

A brief review will be given of the major activities carried on at the headquarters office during the year including membership service, editorial, advertising, sales promotion of publications, committee organization, and general administrative work.

A substantial increase in membership has been recorded through the successful work of the Membership Committee and Chapter cooperation. The present status is: Honorary Members 3, Presidential Members 25, Members 1,356, Associate Members 540, Junior Members 357, and Student Members 91.

Three new Chapters have been organized and some assistance has been rendered by the national officers and the staff in supplying speakers.

The employment service has been an active department involving heavy correspondence and the placement of many members. Members interested in obtaining employment are urged to use this service.

Technical committee activities and editorial work have been carried on effectively by Mr. James and GUIDE sales promotion was successfully done by Mr. Duysters.

The 1937 edition will soon be ready for distribution and this volume will be the largest and most widely circulated of any previous edition.

Two meetings were held in 1936—the 42nd Annual Meeting in Chicago set a record for attendance and the Semi-Annual Meeting at Buck Hill Falls jointly held with the *A.S.R.E.* was exceptionally well attended.

The members of Illinois and Philadelphia Chapters deserve great credit for their work as hosts on these occasions.

During the year there has been an increase in the volume of work in membership, meetings, publications, employment, publicity, and collections and all of this was handled without much additional personnel. Credit is due the members of the headquarters staff for their cooperation and I am glad of this opportunity to publicly express my appreciation.

For the mass of administrative details it might be significant to point out that the number of letters sent out on work exclusive of publications exceeded 20,000, sales promotion campaigns required mailing over 75,000 pieces. More than a carload of paper was used for THE 1937 GUIDE and if the books were placed end to end the distance covered would be a line of books nearly 2 miles long.

The Membership Committee's report will reveal that the increase this year has meant a full time job to carry out all the details of electing 490 members.

The Finance Committee report indicates a business of the Society of nearly $100,000 exclusive of research. The Guide Committee will advise you of a successful year's operations and it can be concluded that the Society is in a flourishing condition and has enlarged opportunities for service to the profession of Heating, Ventilating and Air Conditioning in 1937.

Respectfully submitted,
A. V. HUTCHINSON, *Secretary.*

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The report of the tellers was presented by Secretary Hutchinson as follows:

Report of Board of Tellers

The votes cast for Officers have been tabulated by your Board of Tellers with the following results:

President—D. S. Boyden	595
First Vice-President—E. H. Gurney	595
Second Vice-President—J. F. McIntire	594
Treasurer—A. J. Offner	595
Members of the Council—Three year term:	
J. J. Aeberly	594
M. C. Beman	595
E. O. Eastwood	594
W. A. Russell	595

The votes cast for Members of the Committee on Research have been tabulated by the Board of Tellers with the following results:

Three year term:		
H. E. Adams		595
A. E. Stacey		595
G. L. Tuve		595
J. H. Van Alsburg		593
J. H. Walker		595
W. G. Boales		1
Total number of legal votes cast	595	
Invalid	45	
	640	

Respectfully submitted,
R. H. CARPENTER, *Chairman*,
R. V. SAWHILL,
F. E. W. BEEBE.

President Larson introduced Capt. R. L. Reiss, London, who presented a lantern slide talk on Government Housing Projects in England.

Report of Committee on Constitution and By-Laws

The work of the Committee on Constitution and By-Laws included a review of the By-Laws of several local Chapters and the preparation of an amendment to Article B-II, Section 5 relating to Life Membership.

J. H. Walker, Detroit, chairman of the Guide Publication Committee, presented the following report:

Report of Guide Publication Committee

It is a pleasure to report that THE GUIDE is now on the press and will be distributed to the members in a very few days. This year the policy of the Committee was to completely revise about one-third of the book with the thought that if this were done every year, a completely new GUIDE would be available every three years.

The important changes in the 1937 edition were in the chapters on the Physical and Physiological Principles of Air Conditioning; Humidification, Dehumidification and Water Cooling Equipment; Automatic Control; Sound Control; Air Distribu-

tion; Gravity and Mechanical Warm Air Furnace Systems; Chimneys and Draft Calculations; Drying Systems; and revisions were made in several other chapters.

We revised altogether a total of nine chapters and made minor revisions in possibly nine or ten more. The size of the book has been slightly increased with the total text pages in the 1937 edition numbering 808 as compared with 792 pages for last year. The advertising in the catalog data section shows an increase of 20 per cent from 262 pages to 316 pages. Last year 12,400 copies of THE GUIDE were printed and this year 14,500 copies have been ordered to fill additional requests.

I want to say a word about the suggestions that you submit on the card that is enclosed each year with THE GUIDE. The Committee welcomes suggestions for THE GUIDE, but you may find that your suggestion has not always been adopted. If it has not, it is probably due to one of two things, either you have suggested something that is outside of the scope of THE GUIDE, or else you have asked for something on which there are no data. The Guide Committee cannot go out and create its own data. It has to take material that the Research Committee or the other Society committees have built up, so do not feel hurt if your suggestion has not been embodied in THE GUIDE. It certainly has been given full consideration.

I want to acknowledge the very fine assistance of Mr. James of the Society office. He bore the brunt of the editorial and proofreading work and was of great help to the Committee. The work of the Committee has been made much easier by having Mr. James working on it. I want to express my appreciation to the members of the Committee; they all worked very hard. Mr. Chairman, the Committee presents THE GUIDE.

Guide Publication Committee,
J. H. WALKER, *Chairman.*

Messages of greeting were read from E. H. Gurney, Toronto, first vice-president of the Society, en route home from London from a round the world trip, and from Allen W. Williams, managing director, *National Warm Air Heating and Air Conditioning Association.*

W. T. Jones, Boston, Mass., past president of the Society, said that never before had the Society held a meeting when a disaster of the magnitude of the present flood threatened to destroy cities where members of the Society had homes or business interests. He was presenting the suggestions of several members who believed that some tangible evidence of sympathy for the flooded area be extended through the local St. Louis Chapter. Mr. Jones then presented the following resolution:

That, the representatives of the American Red Cross St. Louis Chapter be permitted to collect any contributions that might be offered by the members assembled at this meeting.

The motion was seconded by T. H. Urdhal, Washington, D. C.

As a result of the collection $162.00 was presented to the St. Louis representative of the Red Cross.

The report of the A.S.H.V.E., *A.S.R.E.* Committee on National Standards for Air Conditioning Applications was presented by John Howatt, Chicago.

Report of A.S.H.V.E.-*A.S.R.E.* Committee on National Standards for Air Conditioning Applications

At the joint convention held last summer, in Buck Hill Falls, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Society of Refrigerating Engineers* were authorized, in the presence of these two Societies, to appoint a Committee to serve jointly in the preparation of national standards or code for air conditioning applications.

The presidents of the two societies, G. L. Larson and L. S. Morse, appointed a Committee consisting of ten. The Committee had its original meeting in Pittsburgh

at which L. A. Harding was chosen as chairman. As President Larson told you, the presentation of this report was to have been made by Mr. Harding, but he is home, ill in bed, and could not attend the Annual Meeting.

At that original meeting in Pittsburgh, a full attendance was had and the tentative draft of standards prepared. The Committee has had several meetings since, two in Buffalo, and each of those meetings resulted in revisions, additions and deletions to the original draft until, today, we have what we believe is a set of standards that is fairly comprehensive and covers in a general way the requirements of comfort air conditioning in North America. However, the work of the Committee is not completed. This fourth revision needs more discussion.

All we can report, today, is that we believe this Committee has progressed a considerable distance in this work, and we know that the work of the Committee will have to be continued, for a part at least, through 1937. The report we are making, then, is a report to you that the Committee is working and making progress and will have to be continued further.

Installation of Officers

W. T. Jones, Boston, past president of the Society, was requested to conduct the installation of officers. President-elect D. S. Boyden, Boston, was introduced by W. H. Driscoll, New York. Second Vice-Pres. J. F. McIntire, Detroit, was introduced by F. B. Rowley, Minneapolis, and A. J. Offner, treasurer, New York, was presented by Mr. Jones.

The new members of the Council, M. C. Beman, Buffalo, W. A. Russell, Kansas City, and J. J. Aeberly, Chicago, were presented by Mr. Jones.

President Boyden assumed the chair and introduced J. E. Hillemeyer, St. Louis, former president of the *National District Heating Association,* who presented greetings from that body.

Resolutions

The report of the Resolutions Committee was presented by G. L. Wiggs, Montreal, who moved the adoption of the following resolutions:

WHEREAS, this, the 43rd Annual Meeting of the Society, has been so eminently successful, not only in the great interest and the valuable contributions of the papers presented by the various authors, and in the splendid entertainment provided by the St. Louis Chapter, but in the delightful surroundings in which the Convention is being held, and especially in the spirit of informality and warm friendship which has been so evident throughout the whole of the meeting,

BE IT, THEREFORE, RESOLVED that the thanks and appreciation of the Society be given:

To each of the *authors* of the various papers for their splendid contributions to the knowledge and literature of the Society,

To the *St. Louis Chapter* and its various committees for the successful way in which they planned, organized and conducted this Annual Meeting,

To the *St. Louis Convention Bureau* for their cooperation and untiring efforts,

To *Dr. A. C. Willard* for his inspiring address at the Annual Banquet,

To the management and staff of *Hotel Statler* for the splendid accommodation and attentive service which they provided,

To the daily and trade *Press* for the attendance of their representatives and for the publicity which they have given to the Meeting,

To *Radio Station KSD* for the time which they so graciously contributed so as to allow one of our Past Presidents to make a broadcast on air conditioning, and

To the *American Rolling Mill Company* for their courtesy in having President Larson as guest speaker on their program in the broadcast over the network of the National Broadcasting Company.

On motion of Mr. Wiggs, seconded by Mr. Russell, the resolutions were adopted.

President Boyden called for any unfinished business and inquired if any new business was to be brought before the meeting. Upon hearing no response, the meeting adjourned at 12:30 P.M.

PROGRAM 43RD ANNUAL MEETING, 1937

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
HOTEL STATLER, ST. LOUIS, MO.
JANUARY 24-27, 1937

Sunday, January 24

10:00 A.M. Council Meeting
4:00 P.M. Committee on Air Conditioning Requirements of Glass, M. L. Carr, *Chairman*

Monday, January 25

9:00 A.M. Registration and Reception, St. Louis Chapter Room
9:30 A.M. Committee on Research—Room 106 (1st Floor)
10:00 A.M. Conference of Chapter Delegates. *Speakers*—President G. L. Larson, W. A. Russell, W. H. Driscoll, D. S. Boyden (Adam Room—17th Floor)
12:30 P.M. Get-Together Luncheon

First Session, Monday, January 25

2:00 P.M. Greeting by C. R. Davis, President St. Louis Chapter
Response by Pres. G. L. Larson
Reports of Officers and Council
Reports of Council Committees
Report of Committee on Research—Lt. Col. W. A. Danielson, *Chairman*
Thermal Properties of Concrete Construction, by F. B. Rowley, A. B. Algren and Robert Lander
The Specific Characteristics of Fans, by M. C. Stuart and J. B. Lusk
Report of Tellers of Election
R. H. Carpenter, *Chairman*
F. E. W. Beebe
R. V. Sawhill

Second Session, Tuesday, January 26

9:00 A.M. Committee on Insulation—L. A. Harding, *Chairman*
9:30 A.M. *Technical Papers:*
Report of Committee on Constitution and By-Laws—R. H. Carpenter, *Chairman*
Report of Guide Publication Committee—J. H. Walker, *Chairman*
A Rational Method of Duct Design, by L. G. Miller
The Noise Characteristics of Air Supply Outlets, by D. J. Stewart and G. F. Drake
Noise in Ventilating Systems and Methods for its Elimination, by J. S. Parkinson

Third Session, Tuesday, January 26

2:00 P.M. *Technical Papers:*
Fever Therapy Induced by Conditioned Air, by F. C. Houghten, Dr. M. B. Ferderber and Carl Gutberlet
Investigations on the Exchanges of Energy Between the Body and Its Environment, by Dr. Charles Sheard and Dr. M. M. D. Williams
Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet
Report of A.S.H.V.E.-*A.S.R.E.* Committee on National Standards for Air Conditioning Applications, by L. A. Harding
4:00 P.M. Meetings of Nominating Committee—Daniel Boone Room (Mezzanine)

XUM

Fourth Session, Wednesday, January 27

9:30 A.M. *Technical Papers:*
Heat Losses and Efficiencies of Fuels in Residential Heating, by R. A. Sherman and R. C. Cross
Performance of Oil-Fired, Warm-Air Furnaces in the Research Residence, by A. P. Kratz and S. Konzo
Condensate and Air Removal Rates in a Vacuum Heating System, by D. W. Nelson
Installation of Officers
Unfinished Business
New Business
Resolutions
Adjournment
12:30 P.M. Council Luncheon and Meeting—Parlor A (Mezzanine)

ENTERTAINMENT EVENTS

Monday, January 25

9:00 A.M. Reception and Registration of Members, Ladies and Guests—Roof Garden, Hotel Statler
12:30 P.M. Get-Together Luncheon for Members, Ladies and Guests (*Obtain Tickets at Time of Registration*)
3-5 P.M. Reception and Tea for Ladies—music and a talk on precious gems
6:30 P.M. Dinner for Past-Presidents—Daniel Boone Room—Mezzanine
7:00 P.M. Dinner for Members, Ladies and Guests—Informal entertainment—Mississippi Levee Levities—a floor show of 10 big acts—music by Al Roth's Orchestra—Ballroom, 16th Floor, Hotel Statler

Tuesday, January 26

10:30 A.M. Ladies assemble for motor trip to Shaw's Garden and the Jewel Box—luncheon at Garavelli's and visit to Jefferson Memorial to see Lindbergh Trophies
7:30 P.M. 43rd Annual Banquet and Dance—Ballroom, Hotel Statler—E. B. Langerberg, Toastmaster. Presentation of Past-President's emblem to Prof. G. L. Larson. *Address*—The Engineer's Place in Modern Society by Dr. Arthur C. Willard, President of University of Illinois. Entertainment by vocal artists and music for dancing by Al Roth and his Columbia Broadcasting Band. Broadcast on N.B.C. Blue-Network Stations at 9 P.M.—Speakers—President Larson and the Armco Ironmaster, Bennett H. Chapple

Wednesday, January 27

2:00 P.M. Inspection of power plant and brewery of Anheuser Busch (*Register for this trip at Registration Desk*)

COMMITTEE ON ARRANGEMENTS

C. R. Davis, *General Chairman*
L. W. Moon, *Vice-Chairman*
C. E. Hartwein, *Finance*
G. W. F. Myers, *Banquet*
R. J. Tenkonohy, *Reception*
P. W. Sodemann, *Transportation*
D. J. Fagin, *Entertainment*
E. E. Carlson, *Publicity*
J. W. COOPER, *Ladies*

No. 1048

THERMAL PROPERTIES OF CONCRETE CONSTRUCTION

By F. B. ROWLEY,* A. B. ALGREN** (*MEMBERS*), AND ROBERT LANDER† (*NON-MEMBER*), MINNEAPOLIS, MINN.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *Portland Cement Association* and the University of Minnesota

THE thermal properties of concrete construction have been under investigation for the past two years as a cooperative research program between the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, the *Portland Cement Association,* and the University of Minnesota. The results of the first part of the work were reported to the Society in January 1936 in a paper entitled, Thermal Properties of Concrete Construction, which was published in A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936.

Since that time the work has been extended to include more masonry walls, insulated monolithic walls, and the thermal conductivity of different aggregates. This paper covers the work completed since the previous report together with a summary and conclusions for the project as a whole.

In order to facilitate comparisons between the results reported in the two papers, the data are presented in the same form for both papers. Insofar as possible the numbers and headlines designating the various tables are identical, and Tables 1, 2, 3, 4, 5 and 6 in this paper may be considered as a continuation of tables of like numbers in the previous paper.

As stated in the previous report, there are several possibilities of improving the thermal properties of concrete construction, among which are the selection of a low conductivity aggregate, the construction of a wall with suitable air spaces, the application of surface finish, and the use of specific insulating materials. The first part of the investigation covered only a limited number of aggregates and did not include any monolithic walls poured with insulation between the slabs. The present report covers a more thorough investigation of different aggregates, both by the hot plate and hot box test methods, addi-

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* Part 2—(Part 1, See Vol. 42).

* Director of Experimental Engineering Laboratories, University of Minnesota.
** Assistant Professor of Mechanical Engineering, University of Minnesota.
† Research Engr., University of Minnesota.

Presented at the 43rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, St. Louis, Mo., January, 1937, by F. B. Rowley.

TABLE 1—MASONRY WALLS—DESCRIPTION AND DATA

WALL NO.	TYPE OF AGGREGATE	DESCRIPTION OF BLOCKS	DENSITY OF MATERIAL IN BLOCKS LB/CU FT	PERCENTAGE OF CORE VOLUME	OVER ALL THICKNESS OF WALL IN INCHES	INSIDE SURFACE FINISH	OUTSIDE SURFACE FINISH	ADDITIONAL INSULATION IN WALL	DENSITY OF INSULATING MATERIAL LB/CU FT	WEIGHT OF INSULATING MATERIAL PER SQ FT OF WALL AREA EXCLUSIVE OF SURFACE FINISH	AGGREGATE USED IN WALL PER CENT BY WEIGHT
9a	**Sand and Gravel**	5″x8″x12″ 2-Core Tile	133.9	49.3	7.78	As Laid	As Laid	None	—	—	72.3 Per Cent Sand 27.7 Per Cent Gravel ⅜ in. to 0
9b	**Sand and Gravel**	5″x8″x12″ 2-Core Tile	133.9	49.3	7.78	As Laid	2 Coats Water Proof White Cement Paint	None	—	—	72.3 Per Cent Sand 27.7 Per Cent Gravel ⅜ in. to 0
11a	**Air-Cooled Slag**	8″x8″x16″ 3-Oval Core	126.3	42.4	7.85	As Laid	As Laid	None	—	—	100 Per Cent Slag ½ in.-0.56.5 Per Cent Fine, 43.5 Per Cent Coarse
12a	**Sand and Gravel**	5″x8″x12″ Blystone Tile	133.7	45.3	7.78	As Laid	As Laid	None	—	—	72.3 Per Cent Sand 27.7 Per Cent Gravel ⅜ in.-0
13a	**Haydite**	8″x8″x16″ 3-Oval Core	67.7	39.6	9.69	As Laid	Furred with Cinder Tile Fig. 5	None	—	—	100 Per Cent Haydite 64 Per Cent Fine 36 Per Cent Coarse
14a	**Haydite**	8″x8″x16″ 3-Oval Core	67.7	39.6	9.69	As Laid	Furred with Haydite Tile Fig. 5	None	—	—	100 Per Cent Haydite 64 Per Cent Fine 36 Per Cent Coarse
2c	**Haydite**	8″x8″x16″ 3-Oval Core	67.7	39.6	7.91	As Laid	2 Coats White Water Proof Cement Paint	Cores filled with Granulated Cork	5.06	1.33	100 Per Cent Haydite 64 Per Cent Fine 36 Per Cent Coarse
3c	**Sand and Gravel**	8″x8″x16″ 3-Oval Core	126.4	39.8	7.88	As Laid		Cores filled with Granulated Cork	5.14	1.35	73 Per Cent Sand 27 Per Cent Pea Gravel
5b	**Cinder**	8″x12″x16″ 3-Oval Core	86.2	40.2	11.88	As Laid	As Laid	Cores filled with Granulated Cork	5.24	2.10	100 Per Cent Cinders ½ in.-0
15a	**Haydite**	8″x12″x16″	76.7	43.1	11.75	As Laid	As Laid	None	—	—	100 Per Cent Haydite 64 Per Cent Fine 36 Per Cent Coarse
15b	**Haydite**	8″x12″x16″ 3-Oval Core	76.7	43.1	11.75	As Laid	As Laid	Cores filled with Granulated Cork	5.60	2.38	100 Per Cent Haydite 64 Per Cent Fine 36 Per Cent Coarse

TABLE 1 (*Continued*)—MASONRY WALLS—DESCRIPTION AND DATA

WALL NO.	TYPE OF AGGREGATE	FINENESS MODULUS OF AGGREGATE			DRY RODDED WEIGHT OF AGGREGATE LB/CU FT		MIX PROPORTION RODDED DRY VOL.	WATER CEMENT RATIO W/C GAL/SACK	28 DAY COMPRESSIVE STRENGTH AIR DRIED					PER CENT ABSORPTION AT 28 DAYS	
									Area of Specimen Sq In.		Total Breaking Load Lb	Breaking Load Lb/Sq In.			
		Fine	Coarse	Combined	Fine	Coarse			Gro.	Net		Gro.	Net	By Wt.	By Vol.
9a	**Sand and Gravel**	3.15	5.74	3.88	113	105	1:7	7.5	91.2	46.3	11,800	1295	2550	5.7	12.2
9b	**Sand and Gravel**	3.15	5.74	3.88	113	105	1:7	7.5	91.2	46.3	11,800	1295	2550	5.7	12.2
11a	**Air-Cooled Slag**	2.29	5.92	3.92	105	79	1:8	7.3	124.1	71.5	113,400	914	1588	8.8	17.9
12a	**Sand and Gravel**	3.15	5.74	3.88	113	105	1:7	7.5	91.5	50.0	124,900	1367	2500	5.7	12.2
13a	**Haydite**	2.76	5.94	3.92	58	43	1:8.5	10.82	125.5	75.75	97,780	779	1289	24.4	25.5
14a	**Haydite**	2.76	5.94	3.92	58	43	1:8.5	10.82	125.5	75.75	97,780	779	1289	24.4	25.5
2c	**Haydite**	2.76	5.94	3.92	58	43	1:8.5	10.82	125.5	75.75	97,780	779	1289	24.4	25.5
3c	**Sand and Gravel**	3.22	5.68	3.91	108	105	1:10	8.92	124.9	75.18	137,440	1100	1829	8.3	16.7
5b	**Cinder**	4.05	—	4.05	63	—	1:8	12.30	188.5	112.8	154,557	820	1369	14.4	19.9
15a	**Haydite**	2.76	5.94	3.92	58	43	1:8.5	7.8	185.9	105.9	136,750	736	1294	16.0	19.6
15b	**Haydite**	2.76	5.94	3.92	58	43	1:8.5	7.8	185.9	105.9	136,750	736	1294	16.0	19.6

tional work on the insulating value of air spaces, and a rather extensive investigation to determine the value of different types of insulating materials when built as integral parts of monolithic concrete walls.

METHOD OF PROCEDURE

The aggregates used in the construction of the various walls were selected and graded as per specifications. A sieve analysis was made and the different aggregates for a given wall were mixed in the required proportion to give the proper fineness modulus. Preliminary tests were made to determine the proper water to cement ratio to give the required slump and strength characteristics. All blocks for masonry walls with the exception of Walls 9a and 12a were made in a standard block machine at the Crown Sidewalk and Block Co.'s plant, Minneapolis, Minn. They were steam cured for 24 hours and then exposed to air for the remainder of the curing period. After the blocks were thoroughly dried they were built into test walls 5½ ft square using ⅜ in. mortar joints between the blocks. The mortar was buttered on the edges but not on the cross partition between the air cells, thus following recommended and standard construction practice for walls of this type. After allowing a sufficient time for the mortar joints to thoroughly dry the walls were tested by the standard hot box method.

For the monolithic walls the aggregates were selected and graded in the same manner as for masonry construction and the walls were built in the laboratory according to specifications. At the time of pouring the walls, 6 in. x 12 in. cylinders were cast which were later tested to give the strength of the concrete. After the walls were thoroughly dried and cured they were tested in the same manner as the masonry walls.

The group of monolithic walls, Nos. 38a to 43a, inclusive, were poured with a sheet of insulating material in the central plane of the wall. For these walls the forms were assembled, a sheet of the required insulating material set in position with reinforcing bars on each side and the aggregate poured and rodded in place. Samples of the insulating materials used in the different walls were selected during the construction of the walls and later tested by the hot plate method to determine their thermal conductivity. The conductivity values for the insulating materials were corrected for the effect of tie rods and these corrected values, together with selected conductivity values for the aggregates, were used to calculate the over-all coefficient of heat transmission for the walls. These values were then compared with the test results.

The thermal conductivities of additional aggregates were determined by building up test samples 24 in. square by 2 in. thick and testing them by the standard hot plate method. These samples were built by the *Portland Cement Association* according to specifications as given in Table 8. Even though these samples were cast between plates with relatively smooth surfaces, it was practically impossible to get a good contact between the surface of the test plate and the specimen. In order to overcome this difficulty sheets of sponge rubber 3/16 in. thick were cemented to both the hot and cold surfaces of the copper test plates. Four thermocouples, each slightly flat, were placed on each rubber surface and held in place by small pieces of adhesive tape at least 2 in.

from the thermocouples. The test plates were clamped sufficiently tight to insure good contact between the plate and the test specimen through the rubber pad. Thirty-six thickness readings were taken in the test area for each specimen and used to compute the conductivities. Sixty-four thickness readings covering the entire area of sample were used to compute the density of the specimens. This procedure proved satisfactory and duplicate tests checked within about 2 per cent.

The water absorption was determined by immersing the sample vertically

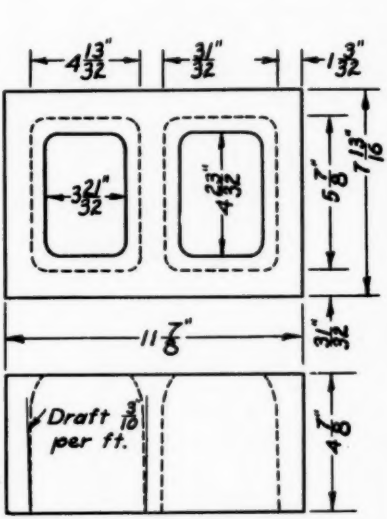

FIG. 1. 2-CORE BLOCK 5 IN. x 8 IN. x 12 IN. USED IN WALLS 9a AND 9b.

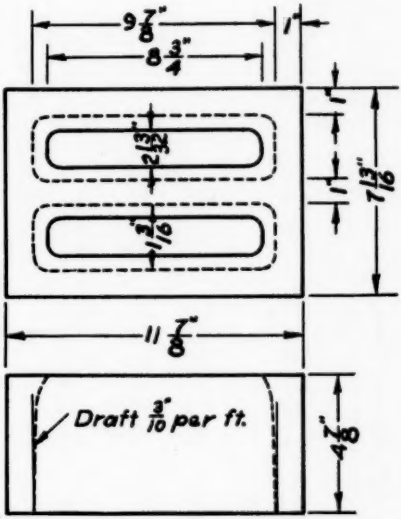

FIG. 2. 2-CORE BLOCK 5 IN. x 8 IN. x 12 IN. USED IN WALL 12a.

in water for 24 hours, then removing and wiping off the surface moisture with a damp cloth and weighing.

The approximate percentage of voids for all but the most porous specimens, 1D to 5D, was obtained by placing the samples vertically in water for 24 hours and then removing and weighing immediately without wiping off the surface water. The gain in weight over the oven dry weight of the sample together with the volume of the sample was used to calculate the percentage of voids. The dry weight of the sample used for calculating the water absorption and approximate percentage of voids was obtained by drying the samples to a constant weight in a steam heated oven at a temperature of approximately 150 F. The approximate percentage of voids for samples 1D to 5D, inclusive, was obtained by a displacement method. In this method a tank 3 in. x 26 in.

TABLE 2—MONOLITHIC WALLS WITH INCLOSED INSULATING MATERIALS—DESCRIPTION AND DATA

WALL NO.	TYPE OF AGGREGATE	DESCRIPTION OF WALL	DENSITY[a] OF CONCRETE IN WALL LB/CU FT	OVERALL THICKNESS OF WALL IN INCHES	INSIDE SURFACE FINISH	OUTSIDE SURFACE FINISH	ADDITIONAL INSULATION IN WALL	DENSITY OF INSULATING MATERIAL LB/CU FT	WEIGHT OF INSULATING MATERIAL PER SQ FT OF WALL AREA	AGGREGATE USED IN WALL, PER CENT BY WEIGHT
38a	Sand and Gravel	4 in. Sand and Gravel Concrete, 1 in. Insulating Membrane, 2½ in. Sand and Gravel Concrete	146.0	7.60	As Cast	As Cast	1 in. Rigid Insulation Board	13.2	1.09	39 Per Cent Sand 61 Per Cent Gravel
38b	Sand and Gravel		146.0	7.60	As Cast	Treated with Fog Spray 1 hour	1 in. Rigid Insulation Board	13.2	1.09	39 Per Cent Sand 61 Per Cent Gravel
38c	Sand and Gravel		146.0	7.60	As Cast	Treated with Fog Spray 19 hours	1 in. Rigid Insulation Board	13.2	1.09	39 Per Cent Sand 61 Per Cent Gravel
39a	Sand and Gravel	4 in. Sand and Gravel Concrete, 1 in. Insulating Membrane, 2½ in. Sand and Gravel Concrete	146.0	7.98	As Cast	As Cast	1 in. Rigid Vapor Proof Insulation Board	22.3	1.92	39 Per Cent Sand 61 Per Cent Gravel
39b	Sand and Gravel		146.0	7.98	As Cast	Treated with Fog Spray 19 hours	1 in. Rigid Vapor Proof Insulation Board	22.3	1.92	39 Per Cent Sand 61 Per Cent Gravel
40a	Sand and Gravel		146.0	7.59	As Cast	As Cast	1 in. Cork Board	8.7	0.72	39 Per Cent Sand 61 Per Cent Gravel
41a	Sand and Gravel		146.0	7.56	As Cast	As Cast	1 in. Water Proof Corrugated Paper	10.0	0.87	39 Per Cent Sand 61 Per Cent Gravel
42a	Sand and Gravel	4 in. Sand and Gravel Concrete, 2 in. Insulating Membrane, 2½ in. Sand and Gravel Concrete	146.0	8.53	As Cast	As Cast	2 in. Insulation made from Shredded Wood and Cement	29.8	4.96	39 Per Cent Sand 61 Per Cent Gravel
42b	Sand and Gravel		146.0	8.53	As Cast	Treated with Fog Spray 19 hours	2 in. Insulation made from Shredded Wood and Cement	29.8	4.96	39 Per Cent Sand 61 Per Cent Gravel
43a	Sand and Gravel	4 in. Sand and Gravel Concrete, 1 in. Insulating Membrane, 2½ in. Sand and Gravel Concrete	146.0	7.22	As Cast	As Cast	1 in. Expanded Aluminum-Magnesium Silicate Insulation	46.6	3.87	39 Per Cent Sand 61 Per Cent Gravel

[a] Note—Results obtained from Test Cylinder of Wall No. 43a.

TABLE 2 (*Continued*)—MONOLITHIC WALLS WITH INCLOSED INSULATING MATERIALS—DESCRIPTION AND DATA

WALL NO.	TYPE OF AGGREGATE	FINENESS MODULUS OF AGGREGATE			DRY RODDED WEIGHT OF AGGREGATE LB/CU FT		MIX PROPORTION DRY RODDED VOLUME	WATER CEMENT RATIO W/C GAL/SACK	SLUMP INCHES	28 DAY COMPRESSIVE STRENGTH			PER CENT ABSORPTION AT 28 DAYS	
		Fine	Coarse	Combined	Fine	Coarse				Area of Cylinder Sq In.	Total Breaking Load Lb	Breaking Load Lb/Sq In.	By Wt.[a]	By Vol.[a]
38a	Sand and Gravel	2.96	6.15	4.91	111.0	98.9	1:2:3½	7.0	7⅜	28.27	121,000	4280	4.85	11.3
38b	Sand and Gravel	2.96	6.15	4.91	111.0	98.9	1:2:3½	7.0	7⅜	28.27	121,000	4280	4.85	11.3
38c	Sand and Gravel	2.96	6.15	4.91	111.0	98.9	1:2:3½	7.0	7⅜	28.27	121,000	4280	4.85	11.3
39a	Sand and Gravel	2.96	6.15	4.91	111.0	98.9	1:2:3½	7.0	7¼	28.27	123,200	4360	4.85	11.3
39b	Sand and Gravel	2.96	6.15	4.91	111.0	98.9	1:2:3½	7.0	7¼	28.27	123,200	4360	4.85	11.3
40a	Sand and Gravel	2.96	6.15	4.91	111.0	98.9	1:2:3½	7.0	7¼	28.14	119,100	4230	4.85	11.3
41a	Sand and Gravel	2.96	6.15	4.91	111.0	98.9	1:2:3½	7.0	7¼	27.76	96,480	3480	4.85	11.3
42a	Sand and Gravel	2.96	6.15	4.91	111.0	98.9	1:2:3½	7.0	7⅜	28.13	97,510	3470	4.85	11.3
42b	Sand and Gravel	2.96	6.15	4.91	111.0	98.9	1:2:3½	7.0	7⅜	28.13	97,510	3470	4.85	11.3
43a	Sand and Gravel	2.96	6.15	4.91	111.0	98.9	1:2:3½	7.0	7¼	28.55	98,480	3440	4.85	11.3
									Average:	7¼	28.19	109,300	3877	

[a] Note—Results obtained from Test Cylinder of Wall No. 43a.

in area and 33 in. deep was equipped with an inclined gage for accurately determining its water level. The tank was then filled with water to a certain level and an amount of water equal to the volume of the specimen was removed. The specimen was immersed in the tank and allowed to stand for 24 hours with the tank covered to reduce evaporation. After the 24-hour period the surface bubbles were removed from the side of the specimen and enough water was added to bring the water in the tank back to the original

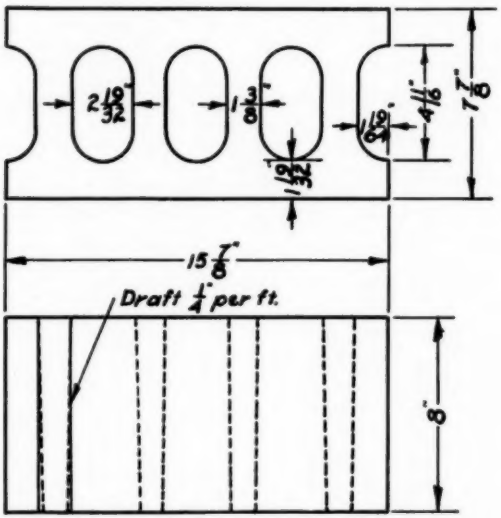

FIG. 3. 3-OVAL CORE 8 IN. X 8 IN. X 16 IN. BLOCK USED IN WALLS 13a, 14a, 2c and 3c.

level. The volume of the water added, divided by the volume of the specimen was used as the approximate percentage of voids. This procedure eliminated the possibility of removing too much water from the surface of the specimen by the standard method.

DESCRIPTION OF MATERIALS TESTED

The complete physical data for the walls tested are shown in Tables 1 and 2 for masonry and monolithic walls, and the construction data are shown in Tables 3 and 4 for masonry and monolithic walls, respectively. Figs. 1 to 4, inclusive, show the detailed dimensions for the blocks used in the construction of the walls. Fig. 5 gives the dimensions for the furring units used on the surfaces of Walls 13a and 14a, and Fig. 6 gives the dimensions for those walls constructed of two monolithic slabs with a sheet of insulating material

between. All blocks, with the exception of those for Walls 9a and 12a were machine made. The blocks of these two walls were made by hand tamping, but as the density was higher and the water absorption slightly less than for those blocks which were machine made from similar materials, the hand blocks were considered satisfactory.

The monolithic walls cast with slabs of insulating materials in the center

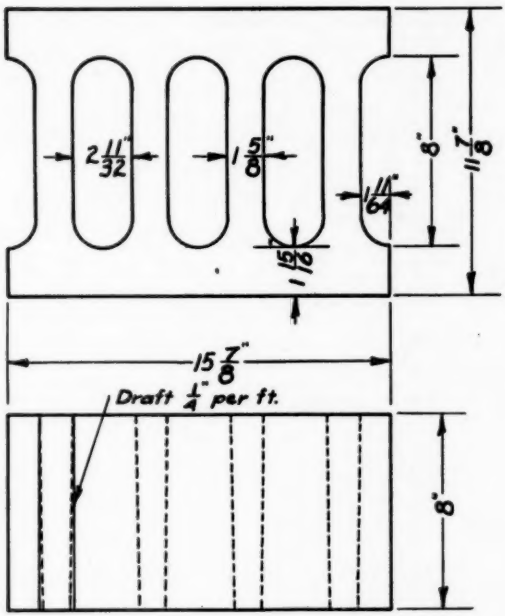

FIG. 4. 3-OVAL CORE BLOCK 8 IN. X 12 IN. X 16 IN. USED IN WALLS 5b, 15a AND 15b.

plane, Nos. 38a to 43a, inclusive, were made by pouring two slabs of concrete one 2½ in. and the other 4 in. thick with a sheet of insulating material as specified between the two slabs. The materials used were Thermax, a material made up of shavings imbedded in cement; Multi-cell, made of several layers of corrugated paper cemented together and water-proofed on the outside; Vermiculite, a mineral expanded by heat and mixed with a cement binder to form a slag; vegetable cork, formed in sheets; and a rigid insulating board made of wood fiber. The density, thickness and thermal conductivity of these various materials are shown in Table 6A.

TABLE 3—CONSTRUCTION DATA FOR MASONRY WALLS

WALL NO.	TYPE OF AGGREGATE IN BLOCKS	AGGREGATE GRADING AND AVERAGE FINENESS MODULUS	BLOCKS MANUFACTURED DATE	WALLS BUILT DATE	INSIDE SURFACE TREATMENT	OUTSIDE SURFACE TREATMENT	DATE OF TREATMENT
9a	**Sand and Gravel**	Average Fineness Modulus = 3.88 Per Cent Passing ½ in. 100.0 No. 16 48.1 ⅜ in. 98.2 No. 30 21.1 No. 4 74.2 No. 50 7.6 No. 8 61.9 No. 100 0.9	11–8–35 to 11–20–35	1–4–36	None	None	
9b	**Sand and Gravel**		11–8–35 to 11–20–35	1–4–36	None	2 Coats Water-Proof White Cement Paint	2–15–36 2–17–36
11a	**Air-Cooled Slag**	Average Fineness Modulus = 3.92 Per Cent Passing ½ in. 100.0 No. 16 40.0 ⅜ in. 93.9 No. 30 27.6 No. 4 63.5 No. 50 19.7 No. 8 51.3 No. 100 12.0	11–22–35	1–3–36	None	None	
12a	**Sand and Gravel**	Average Fineness Modulus = 3.88 Per Cent Passing ½ in. 100.0 No. 16 48.1 ⅜ in. 98.2 No. 30 21.1 No. 4 74.2 No. 50 7.6 No. 8 61.9 No. 100 0.9	11–8–35 to 11–20–35	1–3, 4–36	None	None	
13a	**Haydite**	Average Fineness Modulus = 3.92 Per Cent Passing ½ in. 100.0 No. 16 38.2 ⅜ in. 93.1 No. 30 21.9	11–23–34	1–3–35	None	Furred with Cinder Tile —Fig. 5	1–4–36
14a	**Haydite**	No. 4 68.5 No. 50 14.4 No. 8 61.7 No. 100 10.0	11–23–34	1–3–35	None	Furred with Haydite Tile —Fig. 5	2–8–36
2c	**Haydite**	Average Fineness Modulus = 3.92 Per Cent Passing ½ in. 100.0 No. 16 38.2 ¾ in. 93.1 No. 30 21.9 No. 4 68.5 No. 50 14.4 No. 8 61.7 No. 100 10.0	11–23–34	1–3–35	None	2 Coats Water Proof White Cement Paint	3–16–35
3c	**Sand and Gravel**	Average Fineness Modulus = 3.91 Per Cent Passing ⅜ in. 100.0 No. 30 22.0 No. 4 79.4 No. 50 5.7 No. 8 59.5 No. 100 1.3 No. 16 41.1	11–14–34	1–3–35	None	2 Coats Water-Proof White Cement Paint	3–9–35
5b	**Cinders**	Average Fineness Modulus = 4.05 Per Cent Passing ½ in. 100.0 No. 16 36.8 ⅜ in. 90.4 No. 50 18.2 No. 4 65.1 No. 100 9.5 No. 8 48.1	11–6–35	1–2–35	None	None	
15a	**Haydite**	Average Fineness Modulus = 3.92 Per Cent Passing ½ in. 100.0 No. 16 38.2 ⅜ in. 93.1 No. 30 21.9 No. 4 68.5 No. 50 14.4 No. 8 61.7 No. 100 10.0	5–9–36	6–6–36	None	None	
15b	**Haydite**		5–9–36	6–6–36	None	None	

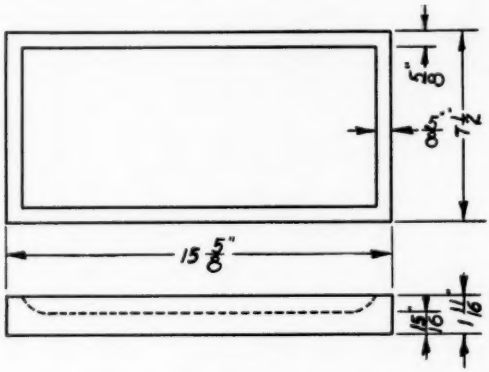

FIG. 5. FURRING UNITS USED ON WALLS 13a AND 14a.

The forms were built up with reinforcing bars laid for each side of the wall. The reinforcing bars consisted of ⅜ in. bars spaced 15 in. horizontally, 18 in. vertically, and tied together at intersections. The reinforcing structure for the two sides of the wall was tied together with ¼ in. tie rods spaced to give one tie rod for each 4½ sq ft area.

TABLE 4—CONSTRUCTION DATA FOR DOUBLE MONOLITHIC WALLS WITH INCLOSED INSULATING MATERIALS

WALL NO.	TYPE OF INSULATION	TYPE OF AGGREGATE IN WALL	GRADING OF AGGREGATE AND AVERAGE FINENESS MODULUS	MIX PROPORTIONS BY VOLUME AND KIND OF MIX	NO. OF LIFTS REQUIRED FOR POURING WALL	DATE OF BUILDING WALL
38a	1 in. Rigid Insulation Board	Sand and Coarse Gravel Concrete	Fineness Modulus of Combined Aggregate = 4.91 Per Cent Passing FINE ¾ in. ..... 100.0 No. 4 ..... 99.7 No. 8 ..... 90.2 No. 16 ..... 70.0 No. 30 ..... 40.0 No. 50 ..... 1.9 No. 100 ..... 1.9 COARSE 1½ in. ..... 100.0 ¾ in. ..... 66.0 No. 4 ..... 14.5 No. 8 ..... 2.0 No. 16 ..... 0.8 No. 30 ..... 0.6 No. 50 ..... 0.4 No. 100 ..... 0.4	1:2:3½ Plastic Mix	4	6–10–35
39a	1 in. Rigid Vapor Proof Insulation Board	Sand and Coarse Gravel Concrete		1:2:3½ Plastic Mix	4	6–11–35
40a	1 in. Corkboard	Sand and Coarse Gravel Concrete		1:2:3½ Plastic Mix	4	6–12–35
41a	1 in. Water Proof Corrugated Paper	Sand and Coarse Gravel Concrete		1:2:3½ Plastic Mix	4	7–30–35
42a	2 in. Insulation made from Shredded Wood and Cement	Sand and Coarse Gravel Concrete		1:2:3½ Plastic Mix	4	8–2–35
43a	1 in. Expanded Aluminum-Magnesium Silicate and Cement Binder	Sand and Coarse Gravel Concrete		1:2:3½ Plastic Mix	4	11–15–35

TABLE 5—TEST DATA AND RESULTS FOR MASONRY WALLS

WALL No.	TEST No.	DATE OF TEST	DESCRIPTION OF WALL	INSIDE SURFACE FINISH	OUTSIDE SURFACE FINISH	ADDITIONAL INSULATION IN WALLS	AIR TEMPERATURE F			COEFFICIENTS OF HEAT TRANSMISSION		
							High Side	Low Side	Mean Temp.	*C*	*U*	*U* Corrected to 15 mph Wind Vel.
9a	48	2-12-36	5 in. x 8 in. x 12 in. 2-Core **Sand and Gravel Tile**	As laid	As laid	None	79.36	0.5	39.93	0.947	0.430	0.541
9b	53	2-25-36	5 in. x 8 in. x 12 in. 2-Core **Sand and Gravel Tile**	As laid	2 Coats Water-Proof White Cement Paint	None	79.97	0.14	40.05	0.899	0.420	0.525
11a	51	2-20-36	8 in. x 8 in. x 16 in. 3-Oval Core **Air Cooled Slag Block**	As laid	As laid	None	79.95	0.16	40.06	0.677	0.364	0.441
12a	50	2-18-36	5 in. x 8 in. x 12 in. **Blystone Tile Sand and Gravel Block**	As laid	As laid	None	79.72	0.39	40.05	0.735	0.380	0.465
13a	47	1-29-36	8 in. x 8 in. x 16 in. 3-Oval Core **Haydite Block**	As laid	Furred with Cinder Tile—Fig. 5	None	80.17	−0.21	39.98	0.343	0.239	0.270
14a	54	2-27-36	8 in. x 8 in. x 16 in. 3-Oval Core **Haydite Block**	As laid	Furred with Haydite Tile—Fig. 5	None	80.84	−0.70	40.07	0.362	0.248	0.281
2c	57	6-19-36	8 in. x 8 in. x 16 in. 3-Oval Core **Haydite Block**	As laid	2 Coats Water-Proof White Cement Paint	Cores Filled with Granulated Cork	80.68	−0.38	40.15	0.199	0.159	0.172
3c	55	6-10-36	8 in. x 8 in. x 16 in. 3-Oval Core **Sand and Gravel Block**	As laid	2 Coats Water-Proof White Cement Paint	Cores Filled with Granulated Cork	80.01	0.4	40.02	0.542	0.321	0.379
5b	56	6-17-36	8 in. x 12 in. x 16 in. 3-Oval Core **Cinder Block**	As laid	As laid	Cores Filled with Granulated Cork	80.58	−0.15	40.22	0.237	0.182	0.199

TABLE 5 (*Continued*)—TEST DATA AND RESULTS FOR MASONRY WALLS

15a	59	7–12–36	8 in. x 12 in. x 16 in. 3-Oval Core **Haydite Block**	As laid	As laid	None	79.96	−0.27	39.85	0.468	0.294	0.342
15b	60	7–15–36	8 in. x 12 in. x 16 in. 3-Oval Core **Haydite Block**	As laid	As laid	Cores Filled with Granulated Cork	80.70	−0.46	40.12	0.168	0.139	0.148

TABLE 5A—INSULATION PLACED IN THE CORE SPACES OF MASONRY WALLS

WALL NO.	DESCRIPTION OF WALL	OUTSIDE SURFACE TREATMENT	INSULATION IN CORE SPACES	DENSITY OF INSULATION LB/CU FT	LB INSULATION/SQ FT WALL AREA	EQUIVALENT THICKNESS OF INSULATION OVER WALL SURFACE	OVER-ALL COEFF. U FOR 15 MILES WIND VELOCITY	OVER-ALL RESISTANCE $1/U$	PER CENT IMPROVEMENT IN OVER-ALL COEFF. U FOR WALLS TESTED	GAIN IN RESISTANCE DUE TO INSULATION	EFFECTIVE CONDUCTIVITY k FOR INSULATING MATERIAL
1a		As Laid	None				0.396	2.52			
1b	8 in. x 8 in. x 16 in.	Painted	None				0.370	2.70	6.66	0.18	
1c	3-Oval Core	Painted	Granulated Cork	5.12	1.34	3.15	0.201	4.97	45.7	2.27	1.38
1d	**Cinder Block**	Painted	Dry Cinders	69.70	18.29	3.15	0.283	3.53	23.5	0.83	3.79
1e		Painted	Rock Wool	14.21	3.72	3.15	0.211	4.74	43.0	2.04	1.54
2a	8 in. x 8 in. x 16 in.	As Laid	None				0.354	2.82			
2b	3-Oval Core	Painted	None				0.354	2.99	5.69	0.17	
2c	**Haydite Block**	Painted	Granulated Cork	5.06	1.33	3.15	0.172	5.81	48.5	2.82	1.12
3b	8 in. x 8 in. x 16 in. 3-Oval Core **Sand and Gravel**	As Laid	None				0.509	1.96			
3c	**Block**	As Laid	Granulated Cork	5.14	1.35	3.15	0.379	2.64	25.5	0.68	4.63
5a	8 in. x 12 in. x 16 in. 3-Oval Core	As Laid	None				0.374	2.67			
5b	**Cinder Block**	As Laid	Granulated Cork	5.24	2.10	4.81	0.199	5.03	46.8	2.36	2.04
15a	8 in. x 12 in. x 16 in. 3-Oval Core	As Laid	None				0.342	2.92			
15b	**Haydite Block**	As Laid	Granulated Cork	5.60	2.38	5.10	0.148	6.76	56.7	3.84	1.33

The properties of the insulating material used in the walls are shown in Table 6A. All of these materials except expanded aluminum-magnesium silicate were purchased from stock. The expanded aluminum-magnesium silicate was built up into slabs 1 in. x 12¾ in. x 33 in., consisting of a mixture of six parts expanded aluminum-magnesium silicate and Vulco aggregate to one part of Lumnite cement. The ratio of expanded aluminum-magnesium silicate to Vulco in the aggregate was 1 to 8 by volume. The Vulco was added to increase the strength. This mixture plus water was spread into a frame and subjected to a pressure of approximately 50 lb per square inch. The resulting slab was smooth on the bottom but somewhat rough on the top surface.

In the walls which were built with vapor proof insulating board, corrugated

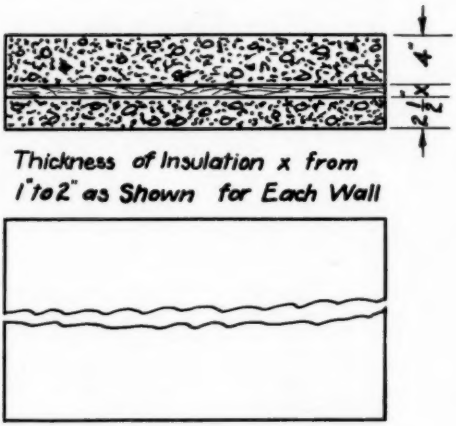

FIG. 6. DETAIL OF MONOLITHIC WALLS WITH INSULATION BETWEEN 2 SLABS

paper, shredded wood and cement insulation, and cork there appeared to be a good bond between the insulating material and the concrete. The wall in which untreated insulating board was used showed a crack between the insulating board and concrete around the edges of the wall, and the wall in which expanded aluminum-magnesium silicate was used showed a 1/16 to 1/8 in. irregular crack through the insulation. These cracks were probably due to the fact that the insulation absorbed water at the time the wall was poured, thus increasing the thickness of the insulation during the setting period for concrete. As the insulation was later dried out it shrunk in thickness and either broke the bond between the insulation and the concrete or else split the insulating material. All walls were dried out for a period of 90 days or more before thermal conductivity tests were made.

The samples built up to determine the thermal conductivity of different aggre-

gates, and shown in Table 8, were divided into two groups. For sand and gravel, limestone, cinders and Haydite the percentages between the fine and coarse aggregates were varied, and also the mix was varied to give different slump tests. For samples 1D to 5D the aggregates were graded to give a fineness modulus of 3.75.

DISCUSSION OF RESULTS

Several interesting comparisons may be drawn from the final test results for the masonry walls as shown in Tables 5 and 5A. Wall 9a, constructed of 5 in. x 8 in. x 12 in. 2-Core sand and gravel tile was first tested with surfaces as laid, and next as wall 9b with outside surface covered with two coats of waterproof white cement paint. The over-all coefficients were 0.541 and 0.525, respectively, showing a reduction of approximately 3 per cent due to coating the outside surface with paint. In the previous series of tests, 8 in. x 8 in. x 16 in. 3-oval core block walls were tested under similar conditions and showed percentages of reduction in the over-all coefficients of 9.5, 8.3 and 3.5, respectively, for cinder, Haydite, and sand and gravel block walls. The results for the sand and gravel block walls check in the two series of tests and the percentage reduction is greater for the more porous aggregates, indicating that it is caused largely by a reduction in surface infiltration losses.

The blocks for Walls 9a and 12a were of the same over-all dimensions and built from the same aggregates. The blocks for Wall 9a shown in Fig. 1 were constructed with two rectangular cores extending between the two surfaces with bridge wall between. The blocks for Wall 12a, shown in Fig. 2, were made up with two long rectangular cores running parallel to the surfaces of the block with bridge wall between. Thus the heat passing from surface to surface of Wall 9a travels across one core space, whereas that in passing through Wall 12a travels across two core spaces. The percentage of core volume was 49.3 for Wall 9a and 45.3 for Wall 12a, but the solid material connecting the two surfaces of Wall 9a was greater than for Wall 12a. This is a good illustration of what may be accomplished by the proper arranging of core spaces. The arrangement for the blocks of Wall 12a as compared with those for Wall 9a reduces the core volume, increases the solid cross-sectional material of the block, and therefore increases its compressive strength, decreases the solid material bridging the two surfaces of the wall, and reduces the over-all heat transfer coefficient by 17 per cent.

Walls 13a and 14a were built by using the 8 in. x 8 in. x 16 in. Haydite block wall, No. 2a, as a foundation and adding cinder furring tile for Wall 13a and Haydite furring tile for Wall 14a. The furring tiles are shown in Fig. 5. The test results for these walls show an over-all conductivity coefficient of 0.270 for Wall 13a, and 0.281 for Wall 14a, or a reduction of 24 per cent and 21 per cent for the two walls, respectively.

The thermal resistance of an air space may often be improved by filling it with some insulating material. The effectiveness of this treatment will depend not only upon the amount and kind of insulation used but on the shape of the air space and the percentage of total heat which may pass around the air space through the solid section of the wall. For two walls of the same

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TABLE 6—TEST DATA AND RESULTS FOR DOUBLE MONOLITHIC WALLS WITH INCLOSED INSULATING MATERIALS

WALL No.	TEST No.	DATE OF TEST	DESCRIPTION OF WALL	INSIDE SURFACE FINISH	OUTSIDE SURFACE FINISH	ADDITIONAL INSULATION IN WALLS	AIR TEMPERATURE F			COEFFICIENTS OF HEAT TRANSMISSION		
							High Side	Low Side	Mean Temp.	*C*	*U*	*U* Corrected to 15 mph Wind Vel.
38a	39	12–14–35	4 in. Sand and Gravel Concrete, 1 in. rigid Insulating Membrane, 2½ in. Sand and Gravel Concrete	As Cast	As Cast	1 in. Rigid Insulation Board	79.99	+0.15	40.07	0.325	0.230	0.258
38b	43	12–31–35		As Cast	Treated with Fog Spray 1 hour. Wall absorbed 12.5 lb Water or 0.5 per cent by Wt. of Wall		80.21	−0.136	40.04	0.325	0.230	0.258
38c	44	1–4–36		As Cast	Treated with Fog Spray 19 hours. Wall absorbed 29 lb Water or 1.16 per cent by Wt. of Wall		80.16	−0.15	40.01	0.321	0.228	0.256
39a	40	12–18–35	4 in. Sand and Gravel Concrete, 1 in. Insulating Membrane, 2½ in. Sand and Gravel Concrete	As Cast	As Cast	1 in. Vapor Proof Rigid Insulation Board	81.19	−1.30	39.95	0.335	0.235	0.265
39b	46	1–10–36		As Cast	Treated with Fog Spray 19 hours. Wall absorbed 32 lb Water or 1.24 per cent by Wt. of Dry Wall		80.20	−0.17	40.01	0.331	0.233	0.262
40a	38	12–10–35	4 in. Sand and Gravel Concrete, 1 in. Cork-Board, 2½ in. Sand and Gravel Concrete	As Cast	As Cast	1 in. Cork Board	80.04	+0.12	40.08	0.360	0.247	0.280
41a	41	12–27–35	4 in. Sand and Gravel Concrete, 1 in. Water Proofed Corrugated Paper, 2½ in. Sand and Gravel Concrete	As Cast	As Cast	1 in. Water Proofed Corrugated Paper	81.15	−1.06	40.05	0.360	0.247	0.280

TABLE 6 (*Continued*)—TEST DATA AND RESULTS FOR DOUBLE MONOLITHIC WALLS WITH INCLOSED INSULATING MATERIALS

42a	42	12–29–35	4 in. Sand and Gravel Concrete, 2 in. Insulation, made from Shredded Wood and Cement 2½ in. Sand and Gravel Concrete	As Cast	As Cast	2 in. Insulation made from Shredded Wood and Cement	80.41	−0.28	40.07	0.377	0.255	0.290
42b	45	1–7–36		As Cast	Treated with Fog Spray 19 hours. Wall absorbed 26 lb of Water or 1.0 per cent by Wt. of Dry Wall		80.11	−0.23	39.94	0.366	0.250	0.284
43a	49	2–14–36	4 in. Sand and Gravel Concrete, 1 in. Expanded Aluminum-Magnesium Silicate and Cement Binder, 2½ in. Sand and Gravel Concrete	As Cast	As Cast	1 in. Expanded Aluminum-Magnesium Silicate and Cement Binder	79.34	+0.47	39.91	1.083	0.456	0.583

TABLE 6A—HOT PLATE TEST DATA AND RESULTS FOR THE INSULATING MATERIALS INCLOSED WITHIN THE DOUBLE MONOLITHIC WALLS OF TABLE 6

WALL NO.	TEST NO.	DATE OF TEST	TYPE OF INSULATION	DENSITY	THICKNESS	TEMPERATURE				C	k
						Hot Side	Cold Side	Temp. Diff.	Mean Temp.		
38a	1182	7–1–35	Rigid Insulation Board	13.2	0.995	113.12	36.30	76.82	74.71	0.344	0.343
39a	1196	11–27–35	Vapor Proof Rigid Insulation Board	22.3	1.034	112.7	37.8	74.9	75.25	0.388	0.401
40a	1194	10–10–35	Corkboard	8.7	0.993	94.0	54.7	39.3	74.35	0.289	0.287
41a	1216	3–13–36	Waterproof Corrugated Paper	10.0	1.046	114.75	35.55	79.20	75.15	0.366	0.383
42a	182[b]	5–19–36	Insulation made from Shredded Wood and Cement	29.8	2.001	91.55	62.90	28.65	77.22	0.385	0.770
43a	1221	6–21–36	Expanded Aluminum-Magnesium Silicate and Cement Binder	46.6	0.997	104.07	34.15	59.92	74.11	1.186	1.182

[b]—24 in. Hot Plate.

TABLE 7—COMPARATIVE VALUES OF COMPUTED AND EXPERI
DOUBLE MONOLITHIC WALLS WITH

WALL NO.	THICKNESS DATA INCHES			THERMAL CONDUCTIVITY k		
	WALL	CONCRETE	INSULATION	CONCRETE[a]	INSULATION	
					HOT PLATE VALUE	CORRECTED FOR TIE RODS
38a	7.60	6.60	0.995	12.0	0.343	0.368
39a	7.98	6.95	1.034	12.0	0.401	0.426
40a	7.59	6.60	0.993	12.0	0.287	0.312
41a	7.56	6.51	1.046	12.0	0.383	0.408
42a	8.53	6.53	2.001	12.0	0.770	0.795
43a	7.22	6.22	0.997	12.0	1.182	1.207

[a]—Assumed.

construction, air space insulation will usually be the most effective for that wall having the lower conductivity aggregate. If the wall is so constructed that the path for heat flow around the air spaces is restricted causing a large percentage of the total heat to flow through the air spaces, then insulation should be effective. If, on the other hand, the wall is so constructed that a large percentage of heat may flow directly through the solid material between the air spaces, then the insulation of the air spaces may do but very little good. In order to effectively insulate a wall all paths of heat flow must be considered. For a masonry wall the solid bridges of concrete between the air spaces will carry the heat past the insulation the same as tie rods will carry the heat through the insulation placed between parallel masonry or monolithic walls. Since it is necessary to have some bond or tie between the two surfaces of most walls this tie should be built with maximum heat resistance when heat resistance is an important factor.

The combined results for several walls tested with and without insulating the core spaces are shown in Table 5A. Part of these walls were reported in the first paper, but are shown here for comparison. The group contains both 8-in. and 12-in. blocks with cinders, Haydite, and sand and gravel as aggregates for the 8-in. blocks, and cinder and Haydite aggregates for the 12-in. blocks.

Four tests were made on the 8-in. cinder block wall, No. 1b without insulation, and Nos. 1c, 1d and 1e having granulated cork, dry cinders, and Rock Wool in the cores for the respective tests. For each other type of construction two tests were run, one without insulation and the other with granulated cork in the core spaces. The fifth from the last column of Table 5A gives the conductivity coefficients corrected to a 15-mile wind velocity over the outside surface. The third from the last column gives the percentage improvement due to insulating the core spaces and shows the important factor in determining whether or not it is worth while to insulate the wall. The last

MENTAL OVER-ALL COEFFICIENT OF HEAT TRANSMISSION FOR INCLOSED INSULATING MATERIALS

COEFFICIENTS OF HEAT TRANSMISSION

f_i	f_o	CALCULATED VALUES		EXPERIMENTAL VALUES		PER CENT DIFF.	
		C	U	C	U	C	U
1.60	1.55	0.307	0.221	0.325	0.230	5.5	3.9
1.60	1.55	0.333	0.234	0.335	0.235	0.6	0.4
1.60	1.55	0.268	0.200	0.360	0.247	25.5	19.0
1.60	1.55	0.322	0.229	0.360	0.247	10.5	7.3
1.60	1.55	0.327	0.231	0.377	0.255	13.3	9.4
1.60	1.55	0.744	0.382	1.083	0.456	31.3	16.2

column gives the effectiveness of the insulating material as used. The values in this column were obtained by assuming that the insulating material used was spread evenly over the full surface of the wall, and then calculating the effective conductivity of this material from the test results for the wall with and without insulation in the core spaces. The high calculated effective values for these materials compared with their actual thermal values is due to the fact that in the test the heat passed around the air spaces regardless of how thoroughly they may have been insulated. A comparison of the results for Walls 1c, 2c and 3c illustrates the fact that the effectiveness of air space insulation is much greater for those walls built of low conductivity aggregates. Thus, granulated cork in a Haydite block gives 1.12, or slightly better than the value of 1.38 when used in a cinder block, but for sand and gravel blocks the effective conductivity is 4.63.

A comparison of the results for the 8-in. and 12-in. masonry walls without insulation in the core spaces shows that for the low conductivity aggregates there is but little difference between the over-all coefficients as obtained for the 8-in. and 12-in. walls. The 8-in. x 8-in. x 16-in. cinder block wall, No. 1a, gave an over-all conductivity coefficient of 0.396 as compared with 0.374 for the 12-in. cinder block wall, 5a. When the outside surface of the 8-in. cinder block wall, 1a, was painted to give Wall 1b, the coefficient was brought down equal to 0.374, or equal to that of the 12-in. wall. The 8-in. uninsulated Haydite wall, 2a, gave a coefficient of 0.357, and the uninsulated 12-in. Haydite wall, 15a, gave a coefficient of 0.342. Two coats of white cement paint on the 8-in. Haydite wall brought the coefficient down to 0.334, which is less than that for the 12-in. wall. When the core spaces of these two walls were filled with cork the conductivity values were 0.172 for the 8-in. wall, No. 2c, and 0.148 for the 12-in. wall, No. 15b.

An inspection of the drawings giving the cross-sectional view of the 8-in. and 12-in. blocks shows the thickness of solid material between the cores of the

TABLE 8—DATA AND TEST RESULTS FOR 24 IN. X 24 IN. X 2 IN. CONCRETE SPECIMENS

PHYSICAL DATA

DESIGNATION ON SAMPLES	TYPE OF AGGREGATE	MIX PROPORTIONS BY VOLUME			SLUMP INCHES	DENSITY OF CONCRETE BONE DRY LB/CU FT	PER CENT ABSORPTION		PER CENT OF VOIDS BY VOL.
		CEMENT	FINE AGGREGATE 0-No. 4	COARSE AGGREGATE No. 4½ IN.			BY WT.	BY VOL.	
S-1-0	Sand and Gravel	1	2.00	2.75	0	144.7	4.7	10.9	11.5
S-2-0		1	2.75	4.50	0	145.7	4.5	10.4	10.9
S-3-0		1	3.50	5.50	0	144.5	4.7	10.8	11.2
S-1-5		1	2.00	2.75	5	142.5	5.9	13.4	13.9
S-1-5		1	2.00	2.75	5	142.5	5.9	13.4	13.9
S-2-5		1	2.75	4.50	5	141.1	6.3	14.2	14.6
S-2-5		1	2.75	4.50	5	141.1	6.3	14.2	14.6
S-3-5		1	3.50	5.50	5	139.2	6.5	14.4	14.7
S-3-5		1	3.50	5.50	5	139.2	6.5	14.4	14.7
L-1-0	Limestone	1	2.00	2.75	0	135.3	7.4	16.0	16.6
L-2-0		1	2.75	4.50	0	137.8	6.7	14.8	15.4
L-3-0		1	3.50	5.50	0	136.4	7.2	15.7	16.3
L-1-3		1	2.00	2.75	3	130.1	9.8	20.4	20.9
L-2-3		1	2.75	4.50	3	126.0	11.4	23.0	23.4
L-3-3		1	3.50	5.50	3	127.3	11.4	23.1	23.4
C-1-0	Cinders	1	2.00	2.75	0	103.6	10.5	17.4	18.2
C-2-0		1	2.75	4.50	0	98.7	12.1	19.1	19.9
C-3-0		1	3.50	5.50	0	92.0	13.8	20.2	21.4
C-1-3		1	2.00	2.75	3	101.4	13.8	22.5	22.8
C-2-3		1	2.75	4.50	3	94.0	17.0	25.6	26.0
C-3-3		1	3.50	5.50	3	94.4	16.0	23.9	24.4
H-1-0	Haydite	1	2.00	2.75	0	80.7	13.5	17.5	18.0
H-2-0		1	2.75	4.50	0	75.0	16.2	19.4	19.8
H-3-0		1	3.50	5.50	0	71.7	18.6	21.3	21.8
H-1-4		1	2.00	2.75	4	78.8	16.5	20.8	21.2
H-2-4		1	2.75	4.50	4	72.4	18.8	21.8	22.2
H-2-4		1	2.75	4.50	4	72.4	18.8	21.8	22.2
H-3-4		1	3.50	5.50	4	71.0	20.3	23.0	23.9
1D	40 Per Cent Expanded Burned Clay, 60 Per Cent Fine Haydite	1	8.00	Fineness Modulus 4.50		57.9	18.2	16.8	18.4
2D	Steam Treated Limestone Slag	1	7.00			74.6	14.7	17.6	27.1
3D	Steam Treated Slag obtained as By-Product of Mfg. of Phosphates	1	8.00			86.6	10.8	15.0	25.5
3DA		1	8.00			91.1	10.5	15.3	21.1
4D	Pumice Mined in California	1	8.00			65.0	20.1	20.9	26.5
5D	Haydite	1	8.50			67.1	17.7	18.9	21.8
5D		1	8.50			67.1	17.7	18.9	21.8

TABLE 8—DATA AND TEST RESULTS FOR 24 IN. X 24 IN. X 2 IN. CONCRETE SPECIMENS

TABLE 8 (*Continued*)—DATA AND TEST RESULTS FOR 24 IN. X 24 IN. X 2 IN. CONCRETE SPECIMENS

					HOT PLATE DATA					
					TEMPERATURES				COEFFICIENTS OF HEAT TRANSMISSION	
DESIGNATION ON SAMPLES	TYPE OF AGGREGATE	TEST NO.	DENSITY AS OF TEST LB/CU FT	THICKNESS INCHES	HIGH SIDE	LOW SIDE	TEMP. DIFF.	MEAN TEMP.	*C*	*k*
S-1-0	Sand and Gravel	143	150.2	1.995	87.51	62.61	24.90	75.06	6.558	13.1
S-2-0		146	150.1	1.990	87.87	61.67	26.20	74.77	6.507	12.9
S-3-0		144	148.1	1.992	88.45	61.55	26.90	75.00	6.606	13.2
S-1-5		147	148.2	1.988	89.28	61.72	27.56	75.50	6.099	12.1
S-1-5		177	148.2	1.988	81.89	67.59	14.30	74.74	6.228	12.4
S-2-5		142	146.4	1.985	86.9	59.7	27.2	73.3	6.256	12.4
S-2-5		153	146.4	1.985	88.79	60.98	27.81	74.89	6.092	12.1
S-3-5		145	144.7	1.982	87.55	61.44	26.11	74.50	6.485	12.85
S-3-5		165	144.7	1.982	86.67	63.64	23.03	75.15	6.293	12.5
L-1-0	Limestone	155	141.0	1.985	90.06	59.68	30.38	74.87	5.670	11.2
L-2-0		152	143.3	1.992	89.50	60.86	28.64	75.18	6.029	12.0
L-3-0		154	141.7	1.996	89.08	60.41	28.67	74.75	5.775	11.5
L-1-3		156	136.3	1.980	89.41	60.31	29.10	74.86	5.294	10.5
L-2-3		150	132.6	1.983	88.28	60.61	27.67	74.45	5.070	10.0
L-3-3		157	133.3	1.983	90.41	60.11	30.30	75.26	4.939	9.79
C-1-0	Cinders	166	110.0	2.038	95.19	55.33	39.86	75.26	2.272	4.63
C-2-0		167	103.9	2.040	97.32	54.11	43.21	75.71	2.106	4.30
C-3-0		168	97.0	2.041	98.09	53.36	44.73	75.72	1.829	3.73
C-1-3		170	110.0	2.028	93.89	56.01	37.88	74.95	2.412	4.89
C-2-3		149	101.9	2.032	99.27	51.14	48.13	75.20	2.156	4.38
C-3-3		169	101.5	2.039	96.48	54.63	41.85	75.55	2.080	4.24
H-1-0	Haydite	158	87.7	2.023	96.87	52.77	44.11	74.82	2.050	4.15
H-2-0		161	81.2	2.034	98.64	52.87	45.77	75.75	1.859	3.78
H-3-0		159	78.4	2.030	96.69	52.94	43.75	74.82	1.810	3.67
H-1-4		163	88.9	2.019	94.68	54.84	39.84	74.76	2.170	4.38
H-2-4		148	80.7	2.017	100.51	50.27	50.24	75.39	1.931	3.89
H-2-4		162	80.7	2.017	96.72	54.26	42.46	75.49	1.913	3.86
H-3-4		160	80.3	2.018	97.16	53.75	43.41	75.45	1.980	4.00
1D	40 Per Cent Expanded Burned Clay, 60 Per Cent Fine Haydite	175	59.9	2.014	93.09	58.05	35.04	75.57	1.134	2.28
2D	Steam Treated Limestone Slag	173	77.5	2.013	94.91	54.06	40.85	74.49	1.128	2.27
3D	Steam Treated Slag obtained as By-Product of Mfg. of Phosphates	172	89.0	2.019	94.11	55.12	38.99	74.62	1.579	3.19
3DA		171	93.4	2.034	92.71	56.15	36.56	74.43	1.683	3.42
4D	Pumice Mined in California	174	69.15	2.024	94.05	55.30	38.75	4.68	1.195	2.42
5D	Haydite	164	70.45	2.024	98.96	52.82	46.14	75.89	1.427	2.89
5D		178	70.45	2.024	87.46	61.90	25.56	74.68	1.391	2.815

8-in. block to be 1⅜ in. and that between the core spaces of the 12-in. blocks to be 1⅝ in. There is, thus, a slightly larger cross-sectional area of solid material connecting the two surfaces of the 12-in. blocks than that connecting the two surfaces of the 8-in. block. The length of path for heat flow through the solid partition of the 8-in. block is 4 $^{11}/_{16}$ in. whereas that for the 12-in. block is 8 in. For the same temperature difference the amount of heat conducted through the solid material for each block would be proportional to the area and inversely proportional to the length of path of solid material, and if there were no heat conducted through the air spaces the heat flow through the 12-in. block would be 0.7 of that conducted through the 8-in. block. Since there is a much greater percentage transferred through the 12-in. block the rate of flow through the air spaces for the 12-in. block must be greater than the rate of flow through the air spaces of the 8-in. block. This increase may be accounted for by the sizes and shape of the air spaces. Since the air spaces are really in the form of narrow slots passing through from surface to surface of the blocks, the solid partition walls bounding these air spaces transfer heat to and from the air in the spaces and actually increase the rate of heat transfer through the air space. This heat transfer from the solid material to the air would normally continue to the central plane of the wall from which point on there would be a transfer from the air to the solid material, and the total amount of heat transferred through the air space would be increased by additional convection currents. The percentage increase in heat transfer would be greater for the longer air space of the 12-in. block than for the shorter air space of the 8-in. block. Thus the air space of the 12-in. block actually conducts more heat from surface to surface of the wall than does the air space of the 8-in. block.

In the first series of tests the use of dry insulating material was investigated when placed between two parallel concrete walls. A 1-in. air space between two masonry walls built up of 4-in. partition tile was filled with Rock Wool and the 2½-in. air space between two monolithic sand and gravel walls was filled first with dry cinders and next with Rock Wool. The results of these tests are shown in Table 10 of the previous report and indicate that when insulation is so used the tie rods or bonds between the two walls must be considered. In the present series of tests six double monolithic walls were built with various types of insulating materials between the slabs. The over-all test results for these walls are shown in Table 6, the hot plate test for the insulating materials in Table 6A, and the summary of the test results on the walls as built compared with calculated results in Table 7. In constructing these walls the area of tie rods was reduced to a minimum, thus giving an effective conductivity for the insulating material only slightly higher than the true value. The last column of Table 7 gives the percentage difference between calculated and test values and shows a substantial agreement for the values of Wall 39a in which a vapor proof insulating board was used. In this case there was a close bond between the insulating material and concrete, and since the board was vapor proof there was no absorption of water to change its thickness or insulating properties. In Wall 38a an untreated insulating board was used which did absorb some water, although in this case the calculated and test values were reasonably close together. For Wall 40a, 1-in. cork board was used, which was rather rough on the surface, permitting the mortar to enter the

board, and thereby decrease its heat resistance. A 19 per cent increase in test value over calculated value shows the effect of this mortar. The insulation in Wall 41a was built up of several layers of corrugated paper having a water proofed surface. In this case the surface of the material was rather soft and the hydraulic pressure of the cement and mortar had a tendency to slightly crush the surface, thereby reducing the effectiveness of the insulation for the finished wall. The insulation, made from shredded wood and cement, used in Wall 42a was of rough and uneven surface, which would allow the entrance of mortar, but in this case the insulating material was 2 in. thick. Therefore the percentage effect due to mortar entering the insulating material was less than for the rough surface of the 1-in. cork. The expanded aluminum-magnesium silicate insulating slab used in Wall 43a absorbed water readily and was materially affected by the wet mortar. These results indicate that if the conductivity of the insulation is corrected for the effect of tie rods, and the surface characteristics of the insulating materials are considered, the over-all coefficients for insulated monolithic wall can be calculated with reasonable accuracy.

The thermal conductivity of monolithic concrete depends largely upon the density of the final concrete, which in turn depends upon the aggregates used, and the amount of water used in the mix. The physical data for aggregates indicate that lower density is obtained with an increase in the water-cement ratio and therefore the thermal conductivity was reduced. The thermal conductivity for six monolithic walls was determined in the first series of tests and the results shown in Table 7 of the first report. The thermal conductivity of nine different aggregates using various mixes was investigated in this series of tests and the results are shown in Table 8. From this table it will be noted that for the same aggregate and for the same slump test there are some differences in density caused by the different grading of the aggregates. These variations in density are, however, not uniform throughout the range. For a given aggregate the conductivities obtained are usually directly proportional to the density. For sand and gravel and limestone aggregates the dry mixes give a denser concrete with higher conductivities. Thus, the dry tamp mix wall, No. 32 of the previous series of tests, gave a conductivity k of 13.1, while the plastic mix, wall No. 31a of the same series, gave a thermal conductivity k of 12.4. The densities were 148.8 and 143.3 respectively. From Table 8 the averages for six sand and gravel samples with zero slump were $k = 13.1$, density 145 lb per cubic foot; and for six samples of 5-in. slump $k = 12.4$, density 141.1 lb per cubic foot. This shows a very close agreement between the conductivity as obtained by the hot plate and hot box method and sets the average value for sand and gravel concrete from 12.4 to 13.1. Three limestone aggregates of zero slump, Table 8, gave an average conductivity of 11.56 at an average density of 136.5 lb per cubic foot, and three samples of 3-in. slump gave an average of 10.08 for conductivity, with 127.8 for density.

Sand and limestone plastic mix wall No. 30a gave a conductivity of 12.1 with a density of 140.3. The fact that the conductivity value for this wall is higher than for these samples tested by the hot plate method is due to the fact that sand was used in the aggregate for the wall, giving a more dense concrete. If the conductivities for the three tests on sand and limestone aggregates are plotted against densities, the results will fall on a straight line. For the cinder and Haydite aggregates the relation between density and plasticity

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of mix is not as definite as with sand and gravel and limestone aggregates and probably is affected more importantly by the degree of compaction of the mix in the mold. For the cinder aggregates, Table 8, the average density of three zero slump samples is 98.1 lb per cubic foot, with a conductivity k of 4.22; and the average of three 3-in. slump samples is 96.6 lb per cubic foot with a conductivity k of 4.45. The density relations are the same as for the other concrete samples, but the conductivities of the samples do not follow the density. This difference is small and might be accounted for by the fact that the surfaces of these samples were rough and test results were more difficult to obtain. For Haydite aggregates the averages for three samples with zero slump were density 75.8, conductivity $k = 3.87$. For four samples of 4-in. slump the averages were density 76.1, conductivity $k = 3.78$. These results are substantially the same as for the cinder aggregates, indicating that for the lower density aggregates such as cinders and Haydite the plasticity of mix is not as important a factor in determining the density and the conductivity as it is for sand and gravel or for limestone aggregates.

ACKNOWLEDGMENT

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DISCUSSION

R. A. MILLER: In Table 1 on concrete, what is the difference between concrete 15a and the concrete block 15b? They were apparently the same but with different deductions.

L. G. MILLER: I would like to inquire about the control of the vapor pressures when the blocks were subjected to the fog spray. Was there any attempt to control the vapor pressure so there would be penetration of vapor, or was the whole block subjected where the vapor pressure was greatest?

PROF. F. B. ROWLEY: In answer to Mr. Miller's question will say that there was apparently a mistake made in transferring the figures from the paper to the lantern slide. The correct figure may be taken from the paper.

In applying the fog spray there was no attempt to control vapor pressure difference across the wall. The wall was placed in the open room, the edges well sealed and the spray applied to one side at substantially room temperature. No measurements were made for condensation through the wall other than the weights taken to measure the water absorbed.

COL. W. A. DANIELSON: In view of the points that came up in our recent insulation meeting about the difficulty of getting contact between the hot-plate and the block, did you experience difficulty?

PROFESSOR ROWLEY: When the slabs of concrete were tested by the hot-plate method a pad of flexible sponge rubber was placed between the surfaces of the concrete block and the test plate. Thermocouples were then placed between the flexible pad and the concrete specimen and the temperatures read by these couples were used in calculating thermal conductivity of the material. The plates were clamped to make a close contact between the rubber pad and material, and also between the rubber pad and plate. The thicknesses of the concrete slab were taken as described in the paper.

No. 1049

THE SPECIFIC CHARACTERISTICS OF FANS

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THE fundamental variables involved in the performance of a given type of fan are quantity, head, efficiency, size and speed. In all problems of fan selection the quantity and head are fixed by a definite external system requirement and the size and speed of the most suitable type are to be determined. The usual quantity-head characteristics, as illustrated in Fig. 1, showing the variation of quantity, head, and efficiency for a given type of fan when the size and speed are fixed, do not lend themselves to the direct solution of problems of this kind.

It is the purpose of this paper to present a new system of fan characteristics, based upon the well-known fan laws which lend themselves to a direct solution of fan selection problems, serve to classify fans as to performance rather than form or design, and are directly useful in the solution of a great variety of fan application problems. These characteristics which are termed *specific characteristics* by the authors, are an outgrowth of the well-known specific speed functions which have been for a long time applied to hydraulic turbines, centrifugal fans and centrifugal pumps.

The basic feature of the specific characteristics of fans, or of other centrifugal machinery such as pumps, hydraulic turbines, propellers, etc., is that they show, for a given type of machine, the variation of size, speed and efficiency when the external system of the machine is fixed. In the case of fans, the specification or fixing of external system fixes the quantity and head of the fan. For fans, then, the specific characteristic would show the variation of size, speed and efficiency for a given type of fan when the quantity and head are fixed. The definitions of the new concepts required, and the explanation of symbols used are as follows:

Notation and Definitions

Q = quantity, thousands of cubic feet per minute.
h = head, inches of water.
e = efficiency, per cent.
D = maximum diameter at tip of blades, inches.
N = speed, revolutions per minute.
V = blade tip speed, feet per minute $= \dfrac{\pi D N}{12}$

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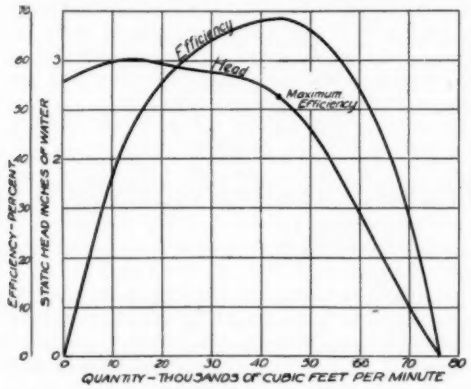

FIG. 1. QUANTITY-HEAD CHARACTERISTICS OF A FAN TYPE D, 72⅝ IN. DIAMETER, 400 RPM

D_s = specific diameter, inches $= D\frac{h^{\frac{1}{4}}}{Q^{\frac{1}{2}}}$

N_s = specific speed, revolutions per minute $= N\frac{Q^{\frac{1}{2}}}{h^{\frac{3}{4}}}$

V_s = specific tip speed, feet per minute, $= \frac{\pi D_s N_s}{12} = \frac{V}{h^{\frac{1}{4}}}$

The *specific diameter*, D_s, of any fan operating at a given efficiency is the diameter of a geometrically similar fan which, when running at its corresponding *specific speed*, N_s, will produce 1000 cfm at 1 in. head at the same efficiency.

The *specific speed*, N_s, of any fan operating at a given efficiency is the speed of a geometrically similar fan of a diameter D_s which will produce 1000 cfm at 1 in. head at the same efficiency.

The *specific tip speed*, V_s, of any fan operating at a given efficiency is the tip speed of a geometrically similar fan with diameter D_s and speed N_s at the same efficiency.

The expressions for specific diameter and specific speed,

$$D_s = D\frac{h^{\frac{1}{4}}}{Q^{\frac{1}{2}}} \text{ and } N_s = N\frac{Q^{\frac{1}{2}}}{h^{\frac{3}{4}}},$$

are derived in Appendix I from the well-known fan laws which state the variation of quantity and head with diameter and speed when operating at a constant efficiency.

The expression for specific tip speed,

$$V_s = \frac{\pi D_s N_s}{12} = \frac{\pi D N}{12 h^{\frac{1}{4}}} = \frac{V}{h^{\frac{1}{4}}}$$

follows from the definition and the expressions for specific diameter and specific speed.

SPECIFIC CHARACTERISTICS

The basis for the computation of the specific characteristics of a fan type is the data from a test of a certain fan of that type as portrayed by the quantity-head characteristic curves. Simultaneous values of quantity, head and efficiency are selected from the curves and the corresponding values of D_s, N_s and V_s are computed.

The details of the computation of the specific characteristics in Fig. 2 from the quantity-head characteristics of Fig. 1 are given in Appendix II. The specific characteristics of Fig. 2 presented in this manner show the functional relations between D_s, N_s, and e in contrast with the functional relations of Q, h, and e as shown by the quantity-head characteristics of Fig. 1. Just as the characteristics of Fig. 1 show variation in quantity and head with speed and size fixed, so the *specific characteristics* of Fig. 2 show variation in size and speed with quantity and head fixed.

From the quantity-head characteristics of Fig. 1, the quantity and head for fans of the same type but of other sizes and speeds may be easily determined by the well understood application of the usual fan laws. In a similar manner, from the *specific characteristics* of Fig. 2, the size and speed for fans of the same type required to produce various quantities and heads may be easily determined by the application of the specific diameter and specific speed coefficients. The details of these determinations are given in another part of this paper. The two curves of Fig. 2 are the only curves actually required to present the entire specific characteristics of a fan type. Other curves between the various specific functions are desirable, however, to develop the

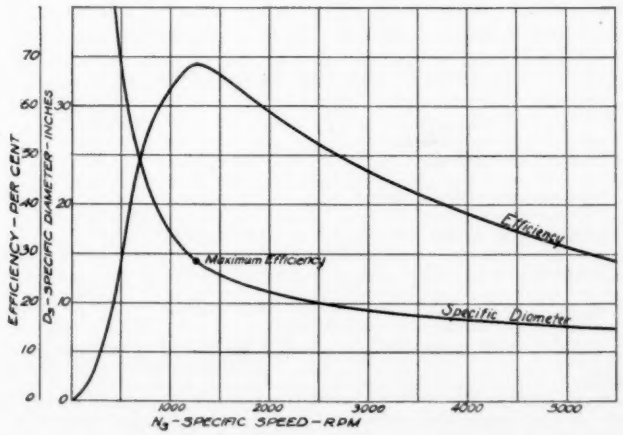

FIG. 2. SPECIFIC CHARACTERISTICS OF A FAN, TYPE D

maximum utility of the specific characteristic method in fan classification, selection and in fan problems. The curves presented in this paper are the following:

a. Specific diameter vs. efficiency, shown in Fig. 3 for a group of 8 fan types.
b. Specific speed vs. efficiency, shown in Fig. 2 for one fan type and in Fig. 4 for a group of fan types.
c. Specific tip speed vs. efficiency, shown in Fig. 5 for a group of fan types, and
d. Specific speed vs. specific diameter shown in Fig. 2 for one fan type and in Fig. 6 for a group of fan types.

The designations of the 8 types of fans for which the specific characteristics were computed and plotted in Figs. 3, 4, 5, and 6 are given in Table 1.

In all problems of fan selection the quantity and head are fixed by a definite external system requirement and the diameter and speed of a fan of the most suitable type are to be determined. From the formulas for specific speed and specific diameter, it is seen that for definite values of quantity and head, the actual diameter of a fan is proportional to its specific diameter and the actual speed is proportional to its specific speed. It is also observed that any quantity and head could be produced by any type fan, within certain limits, if it is the proper size and runs at the proper speed.

Other considerations in the problem which cause limitations are efficiency, actual tip speed, actual speed or actual diameter. A fan is usually selected to operate at or near its best efficiency. This efficiency requirement will limit the corresponding values of D_s and N_s and therefore the values of the actual diameter and speed, for a given type of fan. In many cases the tip speed is limited by the degree of quietness required in operation, and in others by the strength of the parts. The use of synchronous and induction motors directly connected to fans limits the actual speeds at which the fans will operate. Space limitation is the least common restriction placed upon fan selection, but in some cases it does definitely limit the diameter of the fan to be installed.

TABLE 1—FAN TYPE DESIGNATIONS

TYPE	DESCRIPTION	SOURCE
A	Forward curved blade	Fan catalog
B	Forward curved blade, double inlet	Fan catalog
C	Radial blade, double inlet	Fan catalog
D	Backward curved blade	Fan catalog
E	Radial blade, narrow	Fan catalog
F	8 blade airplane propeller fan	Ohio State Engineering Experimental Station, *Bulletin* No. 77
G	Axial flow fan	*A.S.M.E. Transactions* Oct. 1935
H	Slight backward curved blade, narrow	Fan catalog

TABLE 2—BEST EFFICIENCY TABULATIONS FOR 8 FAN TYPES

FAN TYPE	A	B	C	D	E	F	G	H
Best efficiency	67.7	67.2	68.5	68.7	58.0	46.2	70.4	71.5
N_s	900	1180	900	1270	580	4900	2000	310
D_s	11.5	8.7	14.2	14.2	22.0	11.0	15.3	43.0
V_s	2700	2700	3350	4720	3240	14100	8000	3500

When tip speed is a limiting factor, the specific tip speed may be computed from the formula

$$V_s = \frac{V}{h^{\frac{1}{2}}}$$

From Fig. 6 it may be seen that certain fan types are at once eliminated from consideration. The remainder of the problem is to select from the available types, the one which is most suitable as to size, speed and efficiency to perform the required service.

When the speed at which a fan must run is fixed, the specific speed becomes definitely established, because in these cases N, Q and h are all fixed. This fixing of the specific speed limits the choice of fan types to those having good efficiency at the known specific speed. The diameters required of each type can now be computed from the specific diameters corresponding to the specific speed.

When the diameter of the fan is fixed, the specific diameter is fixed because D, Q and h are all fixed. This fixing of the specific diameter limits the choice of fan types to those having good efficiency at the known specific diameter. The speed at which each type must run may be computed from the specific speeds corresponding to the specific diameter. An example in fan selection is given in Appendix III.

THE CLASSIFICATION OF FANS

For a fixed quantity and head the fan speed must be directly proportional to the specific speed and the diameter directly proportional to the specific diameter. Further, specific speed and specific diameter are functions of efficiency, and the values of specific speed and specific diameters at maximum efficiency become significant because of the fixed ratio of the specific characteristics to the actual. The wide variations in the values of N_s and D_s at best efficiency for fans of various types are shown on Figs. 3 and 4. Specific tip speeds are significant at the best efficiency because the actual tip speeds of various fans producing the same head bear the same proportion to their respective specific tip speeds.

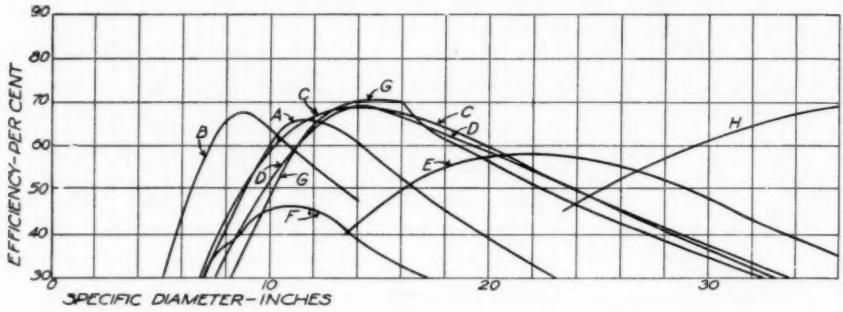

FIG. 3. SPECIFIC DIAMETER VS. EFFICIENCY FOR VARIOUS FAN TYPES

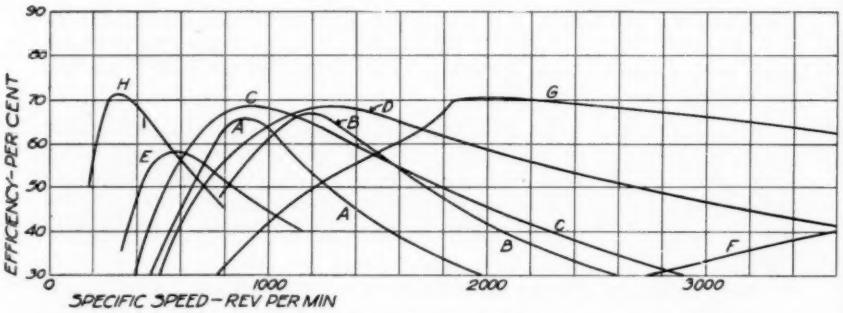

FIG. 4. SPECIFIC SPEED VS. EFFICIENCY FOR VARIOUS FAN TYPES

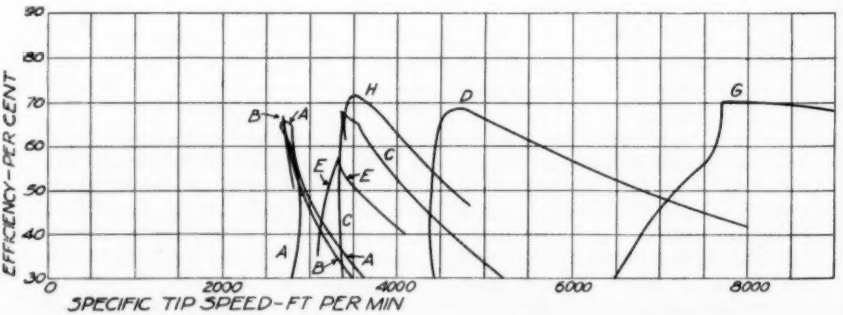

FIG. 5. SPECIFIC TIP SPEED VS. EFFICIENCY FOR VARIOUS FAN TYPES

A tabulation of the values of N_s, D_s and V_s for the 8 fan types at the best efficiency, is given in Table 2.

Fig. 3 which shows the relation between specific diameter and efficiency for several types of fans serves to classify fans according to relative diameter at best efficiency for a given service. Thus fans A, B, and F may be classified as small fans, and E and H as large fans.

Fig. 4 which shows the relation between specific speed and efficiency for several types of fans, serves to classify fans according to the speed at the best efficiency. Fans E and H may be classified as low speed fans, and G and F as

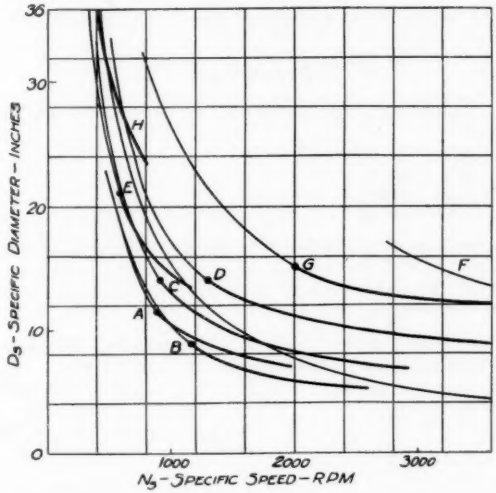

FIG. 6. SPECIFIC SPEED VS SPECIFIC DIAMETER FOR VARIOUS FAN TYPES, SHOWING POINT OF MAXIMUM EFFICIENCY

high speed fans for the same service. Propeller fans, Type F, have extremely high specific speeds.

From the expression $D = D_s \frac{Q^{\frac{1}{2}}}{h^{\frac{1}{4}}}$ it may be seen that fans producing relative large quantities D_s must be small in order to have a small actual diameter. When there is space limitation the fan having the smallest D_s will give the smallest D.

From the expression $N = N_s \frac{h^{\frac{3}{4}}}{Q^{\frac{1}{2}}}$ it is evident that fans operating against relatively high heads must have a low N_s to keep the speed low. For fans producing a large quantity with respect to head, the N_s must be high to keep the actual speed reasonable. For high speed turbine drive N_s must be high and for low speed engine drive N_s must be low for the same service. The diameter is not affected as much by the head as is the speed.

Specific tip speed, V_s, may also be used as a basis for fan classification. The specific tip speed is independent of the quantity requirement as shown by the expression $V = V_s h^{\frac{1}{2}}$. High head fans must have low V_s in order to not have excessive actual tip speeds.

Fans which produce high heads and small quantities must have low N_s and high D_s for the speed to be moderate, such as E and H.

Fans which produce large quantities at low heads must have a high N_s and low D_s such as Fan G.

In addition to their utility in fan classification and selection, the specific characteristics are useful in solving a wide variety of problems in fan application. The methods of the solution of several types of problem are suggested in Appendix IV.

APPENDIX I

DERIVATION OF EXPRESSIONS FOR N_s AND D_s

The fan laws may be expressed as follows:

$$Q = C_1 N D^3 \tag{1}$$
$$h = C_2 N^2 D^2 \tag{2}$$
$$e = \text{constant}$$

To show the variation of N as affected by quantity and head, D may be eliminated from the equations (1) and (2):

$$N = C_3 \frac{h^{\frac{3}{4}}}{Q^{\frac{1}{2}}} \tag{3}$$

When $h = 1$ and $Q = 1$, $N = C_3$; or, C_3 is the speed of a fan required to produce 1 in. head and 1000 cfm. This is the specific speed N_s of the fan.

$$N = N_s \frac{h^{\frac{3}{4}}}{Q^{\frac{1}{2}}} \tag{4}$$

and

$$N_s = N \frac{Q^{\frac{1}{2}}}{h^{\frac{3}{4}}} \tag{5}$$

To show the variation of D as affected by quantity and head, N may be eliminated from the equations (1) and (2):

$$D = C_4 \frac{Q^{\frac{1}{2}}}{h^{\frac{1}{4}}} \tag{6}$$

When $h = 1$, and $Q = 1$, $D = C_4$; or C_4 is the diameter of a fan required to produce 1 in. head and 1000 cfm. This is the specific diameter D_s of the fan.

$$D = D_s \frac{Q^{\frac{1}{2}}}{h^{\frac{1}{4}}} \tag{7}$$

and

$$D_s = D \frac{h^{\frac{1}{4}}}{Q^{\frac{1}{2}}} \tag{8}$$

APPENDIX II

COMPUTATIONS FOR SPECIFIC CHARACTERISTICS

The basis of computations for the specific characteristic of Fig. 2 for fan type D is the data from a test of a 72⅝ in. fan of type D running at 400 rpm.
The items of Table 3 and the source of each are as follows:

Item 1 = Static efficiency, per cent, from curve Fig. 1 corresponding to Q.

Item 2 = Quantity discharged, thousands of cubic feet per minute, selected from curve Fig. 1.
Item 3 = Static head, inches of water, from curve Fig. 1, corresponding to Q.
Item 4 = (Item 2)$^{\frac{1}{2}}$
Item 5 = (Item 3)$^{\frac{1}{2}}$
Item 6 = (Item 3)$^{\frac{3}{4}}$

Item 7 = Specific Speed; $N_s = \dfrac{NQ^{\frac{1}{2}}}{h^{\frac{3}{4}}} = \dfrac{400 \times \text{Item 4}}{\text{Item 6}}$

Item 8 = Specific Diameter; $D_s \dfrac{Dh^{\frac{1}{4}}}{Q^{\frac{1}{2}}} = \dfrac{72.625 \times \text{Item 5}}{\text{Item 4}}$

Item 9 = Specific Tip Speed; $V_s = \dfrac{\pi N_s D_s}{12} = \dfrac{\pi \times \text{Item 7} \times \text{Item 8}}{12}$

APPENDIX III

PROBLEMS IN FAN SELECTION

It is required to select a type, speed and size of a fan to produce at least 25,000 cfm at a head of 4 in. of water at a good efficiency. The tip speed should not exceed 7000 fpm for quietness.

Solution:

$$\text{Maximum specific tip speed, } V_s = \frac{V}{h^{\frac{1}{2}}} = \frac{7000}{2} = 3500 \text{ fpm}$$

From Fig. 5 it will be noted that types D and G are eliminated from consideration because in these types specific tip speed is excessive.

A comparison as indicated in Table 4 of the size, speed and tip speed at best efficiency for various types to give the required quantity and head may be used to select the type. Use the formulas $N = N_s \dfrac{h^{\frac{3}{4}}}{Q^{\frac{1}{2}}}$, $D = D_s \dfrac{Q^{\frac{1}{2}}}{h^{\frac{1}{4}}}$, $V = \dfrac{\pi DN}{12}$, with N_s and D_s determined from curves of Fig. 6.

Type B has the smallest diameter, 30.8 in. All the fans will operate within the limit of tip speed. Types E and H would have relatively large diameters.

If the speed is limited to 600 or 720 rpm

$$N_s = 600 \frac{(25)^{\frac{1}{2}}}{(4)^{\frac{3}{4}}} = 1060 \text{ or } N_s = 720 \frac{(25)^{\frac{1}{2}}}{(4)^{\frac{3}{4}}} = 1270$$

From Fig. 4:

At $N_s = 1060$
- A, 61 per cent efficiency
- B, 64 per cent efficiency
- C, 67 per cent efficiency

At $N_s = 1270$
- A, 50 per cent efficiency
- B, 65.5 per cent efficiency
- C, 62.5 per cent efficiency

Fan C at 600 rpm, from Fig. 3. $D_s = 12.5$, efficiency = 67 per cent.
Fan B at 720 rpm, $D_s = 8.1$, efficiency = 65.5 per cent.

Fan C would have to be much larger than fan B by the ratio $\dfrac{12.5}{8.1}$.

If D were limited to 27 in.

$$D_s = 27 \frac{(4)^{\frac{1}{4}}}{(25)^{\frac{1}{2}}} = \frac{27 \times 1.414}{5} = 7.64$$

From Fig. 3:

Fan B would be the only fan which would give a good efficiency 62 per cent.

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TABLE 3—SPECIFIC CHARACTERISTIC COMPUTATION VALUES

ITEM NUMBER								
1	2	3	4	5	6	7	8	9
e	Q	h	$Q^{\frac{1}{2}}$	$h^{\frac{1}{2}}$	$h^{\frac{3}{2}}$	N_s	D_s	V_s
18.9	5.0	2.90	2.236	1.305	2.223	403	42.4	4470
36.2	10.0	2.98	3.162	1.314	2.270	557	30.2	4400
48.5	15.0	3.00	3.873	1.316	2.281	679	24.7	4390
55.5	20.0	2.96	4.472	1.312	2.257	793	21.3	4420
60.3	25.0	2.92	5.000	1.307	2.235	895	19.0	4450
63.8	30.0	2.89	5.477	1.304	2.218	987	17.3	4470
66.6	35.0	2.83	5.915	1.297	2.183	1083	15.9	4510
68.3	40.0	2.75	6.325	1.288	2.136	1185	14.8	4590
68.7	45.0	2.58	6.710	1.267	2.035	1320	13.7	4730
66.2	50.0	2.29	7.071	1.230	1.861	1520	12.6	5030
61.3	55.0	1.89	7.415	1.172	1.612	1840	11.5	5540
54.1	60.0	1.45	7.746	1.097	1.312	2343	10.3	6300
43.6	65.0	0.96	8.060	0.990	0.970	3320	8.9	7750
27.8	70.0	0.50	8.367	0.841	0.595	5630	7.3	10760
6.5	75.0	0.10	8.660	0.562	0.178	19470	4.7	24000

TABLE 4—COMPARATIVE FAN SELECTION VALUES

FAN TYPE	A	B	C	E	H
Best e	65.7	67.2	68.5	58.0	71.5
N_s	900	1180	900	580	310
D_s	11.5	8.7	14.2	22.0	43.0
N	510	668	510	328	176
D	40.7	30.8	50.2	77.8	152
V	5440	5390	6700	6680	7000

APPENDIX IV

PROBLEMS IN FAN APPLICATION

1. Given type of fan, Q and h. To find size, speed and power required for best efficiency.
   a. From Fig. 4, N_s vs e curve, find N_s at best efficiency.
   b. From Fig. 6, N_s vs D_s curve, find D_s corresponding to N_s for that type of fan.
   c. Solve for N and D from N_s and D_s functions.
   d. Compute power from Q, h, and e $\left(P = \frac{0.1573Qh}{e} \right)$

2. Given type of fan, Q and h. To find diameter, efficiency and power for a given speed.
   a. Compute N_s from N_s function.
   b. From Fig. 4, N_s vs e curve, find efficiency.
   c. From Fig 6, N_s vs D_s, find corresponding D_s.
   d. Solve for D from D_s function.
   e. Compute power from Q, h, and e.

3. Given type of fan, Q and h. To find speed, efficiency and power for a fan with given diameter.
   a. Compute D_s from D_s function.
   b. From Fig. 3, D_s vs e curve, find efficiency.
   c. From Fig. 6, N_s vs D_s, find corresponding N_s.
   d. Solve for N from N_s function.
   e. Compute power from Q, h, and e.

4. Give type, diameter and speed. To find Q and h at best efficiency.
   a. From Fig. 6, N_s vs D_s, find N_s and D_s at best efficiency.
   b. Solve for Q from fan law, i.e. $Q = \left(\frac{N}{N_s}\right)\left(\frac{D}{D_s}\right)^3$
   c. Solve for h from fan law, i.e. $h = \left(\frac{N}{N_s}\right)^2 \left(\frac{D}{D_s}\right)^2$

DISCUSSION

R. L. PERRY[1] (WRITTEN): The method of selecting fans which Professor Stuart and Mr. Lusk have so clearly presented is simple and direct. It can be readily applied in practice by one having limited experience in fan selection. Unfortunately, the specific characteristic data which have been given, have apparently been based upon the *static* head and the volume at the point of maximum *static* efficiency. It would seem desirable, indeed, essential to base the selection upon meeting the *total* head, with the fan operating at the point of maximum *total* efficiency.

Basing the selection upon overcoming static head, and operating at the point of maximum static efficiency tends to conceal the importance of the kinetic energy or velocity head. This results in a failure to meet specifications in some cases, and in the selection of too large a fan with an excessive power requirement in others. When a specification calls only for the delivery of a certain volume against a certain static pressure, it is incomplete in that it does not name the duct area or air velocity if the air is to be delivered to a duct. If a fan exhausting to the atmosphere or through a partition wall is desired, the velocity head at the fan can be recovered by a properly designed conical transition piece, and the static head at the fan can be less than that called for by the specification. Whether the small fan with the transition piece is economically justified by the relative fixed and operating costs, or

[1] Assistant Professor of Agricultural Engineering and Assistant Agricultural Engineer in the Experiment Station, University of California.

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whether it is physically possible to install the transition piece in the desired location are separate questons.

Each type of fan has, at its best operating point, its own particular ratio of static to total head, which will seldom correspond to the ratio occurring in the application. The desirability of using a transition piece is therefore obvious if the fan is to be correctly adapted to its intended use.

For example, let a specification calling for 10,000 cfm to be delivered against a static head of 0.2 in. of water be assumed. Calculation for the propeller fan designated as Type F in Stuart and Lusk's Table 2 results in a diameter of 52 in., a speed of 463 rpm, and a shaft horsepower of 0.68. For 10,000 cfm discharged through a 52 in. circle, the velocity head, not mentioned in the specification, is 0.0286 in. of water. If this fan were called on to deliver air into a duct where its velocity would be 1600 ft per minute, the velocity head at the duct would be 0.16 in. of water, and the total head 0.16 plus 0.2 or 0.36[2] in. of water. The selection made above develops a total head of only 0.0286 plus 0.2 or 0.2286 in. of water, and the 52 in., 463 rpm fan would not do the work.

From Brown's curve[3] for this fan, from which Stuart and Lusk's data were derived, a set of specific characteristics based upon the total head and the volume delivered at the point of maximum total efficiency can be computed; namely, total efficiency = 57 per cent, specific diameter = 8.06, specific speed = 7370. A fan selected from these specific data for delivery of 10,000 cfm against a total head of 0.36 inches of water would have a diameter of 33 in., a speed of 1100 rpm and a shaft horsepower of 1.0. This selection of diameter and speed gives the minimum horsepower requirement for a fan of this type, meeting the assumed conditions.

If the velocity, not mentioned in the assumed specification, were unimportant, as might be the case for delivery through a partition, a transition piece might be used to convert the velocity energy at the fan into static head. Assuming no loss in conversion, and no final velocity head, the fan need meet a total head of only 0.2 in. Using the total head specific characteristics, the diameter is found to be 38 in., the speed 696 rpm and the shaft horsepower 0.553. This is a smaller fan with a lower power requirement than was found above by the use of static head specific characteristics. It does, however, require the transition piece.

It is idle to contend whether the transition piece is a part of the fan or a part of the installation in which the fan is to be placed. The transition piece must be designed for each job, because its inlet dimensions depend directly upon the fan dimensions, while its outlet depends upon the installation.

Now that Stuart and Lusk have shown the simplicity and utility of the method, it is to be hoped that more fan data giving *specific characteristics* will be made available.

J. R. WESKE[4] (WRITTEN): The writer regards this paper as a vital and needed contribution to fan engineering. In his opinion, the relations between the factors of

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[2] Select for 0.36 in. static head, assuming all velocity head lost (no transition piece):

$$D = D_s \frac{Q^{\frac{1}{2}}}{h^{\frac{1}{4}}} = 11 \frac{10^{\frac{1}{2}}}{0.36^{\frac{1}{4}}} = 11 \frac{3.16}{0.775} = 45 \text{ in.}$$

$$N = 4900 \frac{0.36^{\frac{3}{4}}}{10^{\frac{1}{2}}} = 4900 \frac{0.47}{3.16} = 730 \text{ rpm}$$

$$AHP = Qh = 10 \times 0.36 \times 0.157 = 0.565$$

$$SHP = 0.565 \div 0.462 = 1.22$$

[3] The Performance of Propeller Fans, A. I. Brown (Ohio State University Engineering Experiment Station Bulletin 77, p. 25, 1931).

[4] Assistant Professor of Aerodynamics, Case School of Applied Science, Cleveland, Ohio.

design and the performance data of different types of fans are of great practical value equalling in importance the relations of dynamic similarity embodied in the fan laws. While the latter have become indispensable in fan work, progress in the comparative study of different fan types has lagged behind. One reason, the writer suspects, is that there has not been available a theoretical structure to supplement the results of the analysis of empirical data. In the case of centrifugal fans, for example, there exists no precise theory permitting complete analysis of the factors affecting performance. Fortunately, for axial flow fans a satisfactory theory has been developed by C. Keller.[8]

In the design and application of axial flow and centrifugal fans the writer has found specific factors of the type proposed by the authors most useful. As an example, Fig. A is presented showing different fan types arranged according to their specific speed at best efficiency. Fans numbers 1, 2, 3, and 6 were tested by the writer; numbers 4, 7, and 8 were designed by C. Keller. Fan number 5 is identical with fan G analyzed by the authors. The fans are drawn to the same

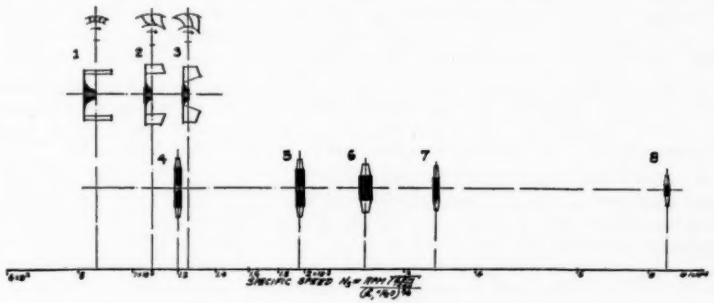

FIG. A. FAN TYPES SHOWING SPECIFIC SPEED IN RELATION TO EFFICIENCY

scale, the specific diameter serving as a basis for comparison. It can be seen from the figure that the wheels increase in size as the speed decreases. This tendency can be partly offset by proper choice of the width and the angle of the blades.

The authors make the statement that fans are generally operated at, or near, their best efficiency point, meaning thereby the maximum efficiency of the individual fan characteristic. In addition to this efficiency, a best attainable "type efficiency" may be defined, which varies with the specific speed of the type. In the case of axial flow fans the value of this type efficiency may be determined by calculation. It has been the writer's experience that some fans are designed for a lower efficiency than this best type efficiency, chiefly for two reasons: either the fan is dimensioned to handle an excessive quantity of air for a given speed, or the tip speed of the fan is reduced below its optimum value in order to decrease the noise. Whenever economy plays a role in such cases it is advisable to determine the loss of type efficiency empirically by means of a chart similar to Fig. 4 of the paper or, where possible, theoretically.

S. H. DOWNS: I haven't any prepared discussion on this paper, although I have participated in some of the work on fans that Professor Stuart has been doing in connection with the *A.S.M.E.* Power Test Code Committee. I believe Professor Stuart's analysis, as given in the paper, is a very useful thing.

[8] C. Keller, Axialgeblaese, Zuerich, 1934.

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As mentioned, this type of analysis has been used quite extensively on machines for handling water, but it is readily applicable to machines handling any other fluids such as air or other gases. The considerations brought out in this paper, however, point to a maximum single point of efficiency, although that is not the way that fans are usually applied. All commercial fans are designed to give a very broad peak on the efficiency curve rather than coming to a sharp point. It is possible in certain applications to choose a fan according to this method to hit the peak of efficiency, but you might also choose another size a little off the peak and actually take less horsepower to do the same type of work on a given installation.

The method, therefore, would not be actually foolproof and after the particular choice had been made on any given system, you would still have to apply the general rules of applications to see that you had picked out a size of fan that was actually available on the market, and also that you did not fool yourself a little bit in trying to get a maximum of efficiency and thereby a little more horsepower than if you had striven for a fan of lower velocity head at the outlet that might give you the result of lower horsepower. This would be of particular importance in the type of fan that has specific characteristics so that the motor is fully loaded at all times. In some cases this ignoring of lowest horsepower and aiming at highest efficiency might throw you to a point where your motor is just a little bit heavily loaded.

ERNEST SZEKELY: I don't think I have any comment except what Mr. Downs made, but may emphasize that the practical limitations would prevent us from using the method because, of course, we have to use sizes available rather than specific diameters that may be obtained from the curve. That is all I could say.

M. C. STUART AND J. B. LUSK: Professor Perry's discussion points out that in cases where the discharge kinetic energy is of importance, the specific characteristics should be based on total head, since the static head characteristics alone give no information as to the discharge velocity head. With this the authors readily agree. The purpose of this paper was primarily to present the general method of specific characteristics of fans, and it appears more convenient to base the method on static head only.

Just as both total and static head characteristics are necessary to show completely the performance of a given fan, so it is necessary to show both total and static specific characteristics of a fan type to render the method completely useful in all cases. It must be considered that in some cases static head only is of interest in a fan installation, and even when transition or diffusers are used, there are considerable losses in the conversion to static pressure.

We find Professor Perry's carefully worked out examples correct and very instructive in the use of the method. His discussion is a good illustration of the ease with which the method may be used and a careful study of his discussion in connection with the paper will repay anyone interested.

Dr. Weske adds some very valuable data pertaining to the specific characteristics of fans of various types. It should be added that the work of Keller referred to by Dr. Weske is now available in English translation by Professor Marks.[6]

The authors agree with Mr. Downs that design should not necessarily be at the highest efficiency as shown by the specific characteristic curves. It must be emphasized that the method is a convenient aid, but not a substitute for sound engineering, in the problems of fan selection and application.

In reply to Mr. Szekely, it may be said that the necessity for use of standard sizes does not preclude the use of the method, because, in fact, it will determine the nearest standard size of a type which should be used, and then further application of this method will give the complete characteristics of the fan of diameter chosen.

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[6] Keller-Marks; Axial Flow Fans.

No. 1050

A RATIONAL METHOD OF DUCT DESIGN

By L. G. MILLER* (*MEMBER*), EAST LANSING, MICH.

ANYTHING rational has *reason or understanding* according to Noah Webster. The following method of duct design is presented because each step is a result of reasoning. This outline is intended to serve only as a guide for the method. No claim is made for the accuracy of the data, but it is hoped that those having data known to be authentic will present it and that properly equipped laboratories will supply that information needed to satisfy controversial views.

Before the problem of air duct design can be approached, such items as design and location of grilles, quantity and quality of air to be circulated, and the location of the fan and conditioning apparatus will have been determined by engineering calculations and estimates. An illustrative problem is interspersed throughout the discussion of the method. The results of the engineering calculations as they apply are included in Table 1.

DESIGN OF DUCT SYSTEM

The first step in actual design of a duct system is to connect the grilles with the fan and other apparatus into a workable closed system. It is hardly necessary to state that the plans, or actual structure, must be thoroughly inspected for structural details that might hinder or aid in erecting the duct system. Economy of construction and of operation, two diametrically opposed considerations, should be weighed one against the other. In order to facilitate such studies it seems desirable to add to the Symbols for Heating, Ventilating and Air Conditioning Drawings given in Chapter 44 of THE A.S.H.V.E. GUIDE 1937 some items commonly encountered in duct layout. A partial list appears in Fig. 1. Using these symbols, sketch on the plan involved, a line layout. Such a sketch for the flow ducts in a residence is illustrated by Fig. 2.

Selection of Design Branch Duct

Select by inspection the duct appearing to have the least unavoidable resistance and design it for the highest usable velocity. Outlet No. 4 of Fig. 2, requiring that 145 cfm be delivered from the grille at 300 fpm velocity, is selected because it is the shortest run and appears to have little natural re-

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Presented at the 43rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, St. Louis, Mo., January, 1937.

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sistance. Some work is being done to reduce noise generation and transmission by the duct systems. Findings to date with more definite recommendation should be published in subsequent editions of THE A.S.H.V.E. GUIDE to allow higher velocities and resultant smaller ducts. Based upon data given in THE

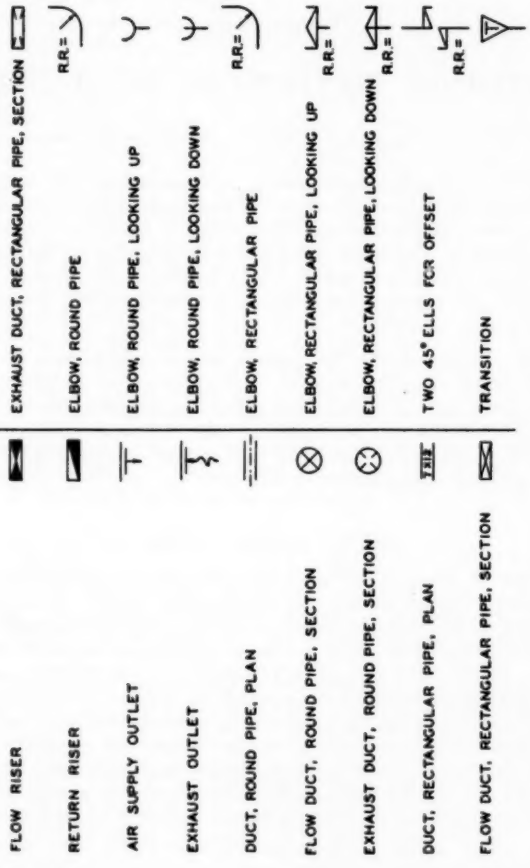

FIG. 1. SYMBOLS FOR HEATING, VENTILATING AND AIR CONDITIONING DRAWINGS

A.S.H.V.E. GUIDE 1937, the highest velocity which should be used in this vertical wall stack is 750 fpm and in the branch horizontal duct 1000 fpm. The cross-sectional area of these sections would therefore be $\frac{145}{750} = 0.193$ sq ft = 27 sq in. and $\frac{145}{1000} = 0.145$ sq ft = 22 sq in. A 3 x 9 in. wall stack.

having a cross-sectional area of 27 sq. in. would be right. The table given on p. 372 of THE A.S.H.V.E. GUIDE 1937, Circular Equivalents of Rectangular Ducts for Equal Friction, should be extended to include at least 3 and 3½ in. dimensions to accommodate designers of residential heating. The equivalent of a 3 x 9 in. duct is 5.5 in. One dimension of a trunk line system is usually determined by structural limitations. In this particular example, head room in

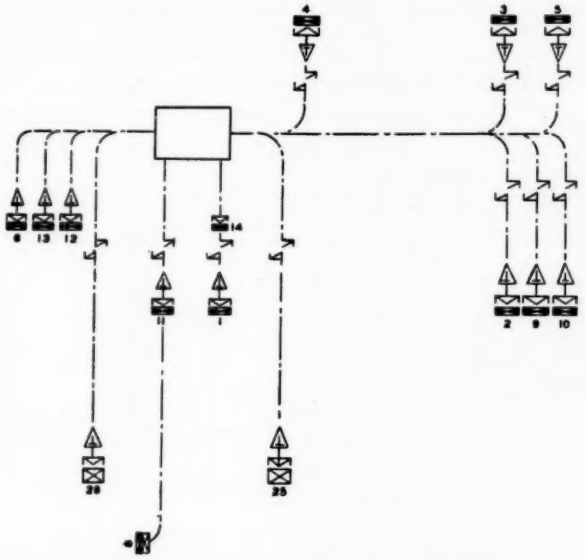

FIG. 2. RESIDENTIAL AIR FLOW LINE DRAWING LAYOUT

the basement and the limitations of depth to that of 8 in. commercial lumber joist, dictates a uniform duct depth of 7 in. The branch duct leading to the riser of grille No. 4 would be $\frac{22}{7}$ say 3 in. in width, equivalent to 4.8 in. diameter.

Branch Duct Resistance

A considerable amount of the resistance in a system is offered by the elbows, takeoffs, transitions, and other fittings. A curve of average values of percentages of velocity head lost in elbows with various degrees of curvature is shown in Fig. 3. Much more work should be done under various conditions of velocity, shape, size, etc. to indicate whether one such curve will suffice or if a group of curves would be required before the data can be used with commercial

accuracy. It is obvious that nothing is to be gained by using elbows with a *radius ratio* greater than 2. Space limitations, manufacturing costs, use of standard fittings and many other items have a part in the actual degree of curvature adopted in each case. Where unusual circumstances do not dictate, the *radius ratios* selected for economy will not vary greatly from unity. Such an elbow would cause a drop in pressure equal to 22.0 per cent of the velocity head (Fig. 3) and with 135 F air, traveling 750 fpm, the actual drop in this elbow would be 0.031 in. w.g. × 0.22 = 0.0068 in. A stack head as ordinarily installed has a square throat, this is a *radius ratio* of 0.5, for which Fig 3

TABLE 1—RESIDENTIAL AIR QUANTITIES REQUIRED

ROOM	GRILLE NO.	CFM	DELIVERED VELOCITY FPM
Living Room 1	1	95	500
Living Room 2	2	95	500
Dining Room	3	100	425
Sun Room 1	4	145	300
Sun Room 2	5	145	300
Kitchen	6	60	300
Hall Down	8	100	300
N. W. Bedroom	9	130	500
S. W. Bedroom	10	100	400
N. E. Bedroom	11	125	500
S. E. Bedroom	12	115	300
Hall Up	13	50	100
Bath	14	45	200
Recreation 1	25	125	100
Recreation 2	26	125	100

shows 105 per cent velocity head. The resistance is equivalent to 0.031 × 1.05 = 0.0325 in. w.g.

OFFSETS AND TRANSITIONS

It is often necessary to change the alinement of a duct to clear some portion of the structure or to reach a grille or riser. Two 45 deg ells, connected with enough pipe to provide the necessary offset, are usually employed. Such a combination of ells will not exceed in resistance that of a single 90 deg ell and can be estimated as 0.0068 in. w.g. It should be noted that all combinations of ells cannot be so easily estimated and the designer should have more complete data available.

Branch take-offs are designed in such a variety that it is impossible to attach any great weight to the values of loss in pressure due to these fittings as used in the illustrative problem. Many engineers use a combination approximating two 45 deg ells. A simple scheme illustrated in Fig. 4 is suggested. This branch take-off functions equally well for parallel and 90 deg directions and with slight modifications for other angles. In pressure drop, it is almost identi-

cal with one 90 deg ell and will be so estimated in this discussion, i. e., equal to 0.0068 in. w.g.

The loss through the transition fitting from the 3 x 7 in. branch duct, with the larger dimension vertical, to the 3 x 9 in. elbow with larger dimension horizontal, will of course depend upon the abruptness of the fitting. Very little data are available concerning the resistance of these fittings, estimated generally from 0.25 to 0.85 of the velocity head. Since in this illustrative problem the

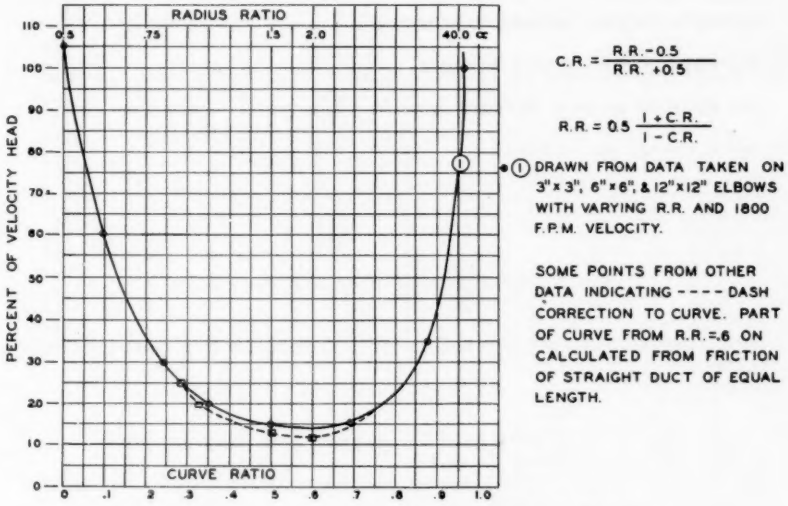

FIG. 3. PERCENTAGES OF VELOCITY HEAD LOST IN ELBOWS

drop from 1000 to 750 fpm makes available the difference in velocity head equal to 0.025 in. w.g. which is 0.45 of the velocity head, it may be assumed that this gain will offset the loss, a fact no doubt very nearly the truth.

DUCT FRICTIONAL RESISTANCES

To the drop in pressure in the fittings must be added that due to friction in the straight duct. This drop may be determined algebraically by a modification of the Babcock-Unwin formula

$$p = 0.0358 \frac{flv^2r}{d_e} \quad \text{.....[1]}$$

where

p = pressure drop, inches water gage.
f = friction factor.
l = length of duct, feet.
v = velocity, feet per second.
r = density, pounds per cubic foot.
d_e = diameter of a circular pipe or the equivalent diameter of a rectangular pipe, inches.

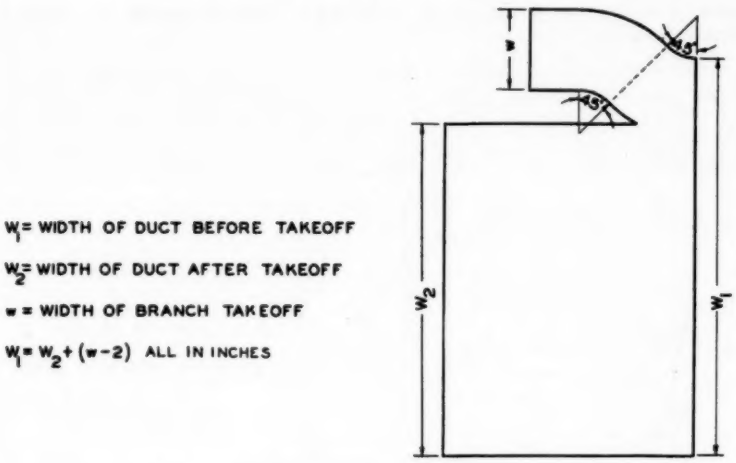

FIG. 4. DETERMINATION OF BRANCH DUCT SIZES

To illustrate, check the drop in 7 feet of wall stack 3 x 9 in. leading to grille No. 4:

$l = 7$ ft.
$v = 12.5$ fps.
$r = 0.067$ for air at 135 F containing 45 grains of moisture per pound.
$d_e = 5.5$ in. as previously determined.
$f =$ value determined from Fig. 5 by calculating the Reynolds Number, (R_e).
$R_e = \frac{Dv}{k}$ where k is the kinematic viscosity. From Fig. 6 the kinematic viscosity of the air in the flow risers is 20×10^{-5}.
$D = \frac{d_e}{12} =$ equivalent pipe diameter, feet.
$K = \frac{\mu}{\rho} =$ kinematic viscosity.
$\mu =$ absolute viscosity, pounds mass per foot second.
$\rho =$ density, pounds mass per cubic foot.

$$R_e = \frac{0.46 \times 12.5}{20 \times 15^{-5}} = 28{,}750$$

The table on Fig. 5 recommends the use of curve 4 for a sheet metal duct of this size. From this curve the value of

$$f = 0.026.$$

From equation (1),

$$p = 0.0358 \frac{0.026 \times 7 \times (12.5)^2 \times 0.067}{5.5} = 0.0123 \text{ in. w.g.}$$

The summary of the losses in this branch of duct length appears in Table 2.

To be sure that velocities and sizes will fall within reasonable limits it is desirable next to check the system which appears upon inspection to have the greatest natural resistance. The duct leading to grille No. 10 is the longest since it leads to a second floor room. It carries 100 cfm to be delivered at 500 fpm.

It is convenient to take advantage of the fact that since the resistance of both duct and fitting varies approximately with the square of the velocity the resistance of one may be expressed in terms of the other. The total resistance of the line leading to grille No. 4 in terms of an equivalent length is given in Table 3.

The duct leading to grille No. 10, estimated in the same manner, results in

TABLE 2—SUMMARY OF HEAD LOSSES TO GRILLE NO. 4

FITTING	VELOCITY	VELOCITY HEAD	PER CENT HEAD LOST	HEAD LOSS IN. W.G.
Stack head	12.5	0.031	105	0.0325
Elbow to stack	12.5	0.031	22	0.0068
Offset	$16\frac{2}{3}$	0.055	22	0.0121
Transition	$16\frac{2}{3}$	0.055	0	0.0000
Branch take-off	$16\frac{2}{3}$	0.055	22	0.0121
Riser	12.5	0.031	...	0.0123
Total				0.0758

TABLE 3—TOTAL RESISTANCE OF DUCT LEADING TO GRILLE NO. 4

FITTING	RADIUS RATIO	WIDTHS OF PIPE OFFERING EQUAL RESISTANCE	WIDTH FT	EQUIVALENT LENGTH FT
Stack head	0.5	47.5	3/12	11.87
Elbow to stack	1.0	9.0	3/12	2.25
Offset	1.0	9.0	7/12	5.25
Branch take-off	1.0	9.0	3/12	2.25
Actual duct	...	....	....	7.00
Total				28.62

TABLE 4—DESIGN SUMMARY OF HEAD LOSSES TO GRILLE NO. 4

FITTING	VELOCITY	VELOCITY HEAD	PER CENT HEAD LOST	HEAD LOSS IN. W.G.
Stack head	480	0.0127	105	0.0133
Elbow to stack	480	0.0127	22	0.0028
Offset	687	0.026	22	0.0057
Branch take-off	687	0.026	22	0.0028
Riser (16.0 ft)	480	....	...	0.0119
Branch duct (6.5 ft)	687	....	...	0.0120
Total				0.0485

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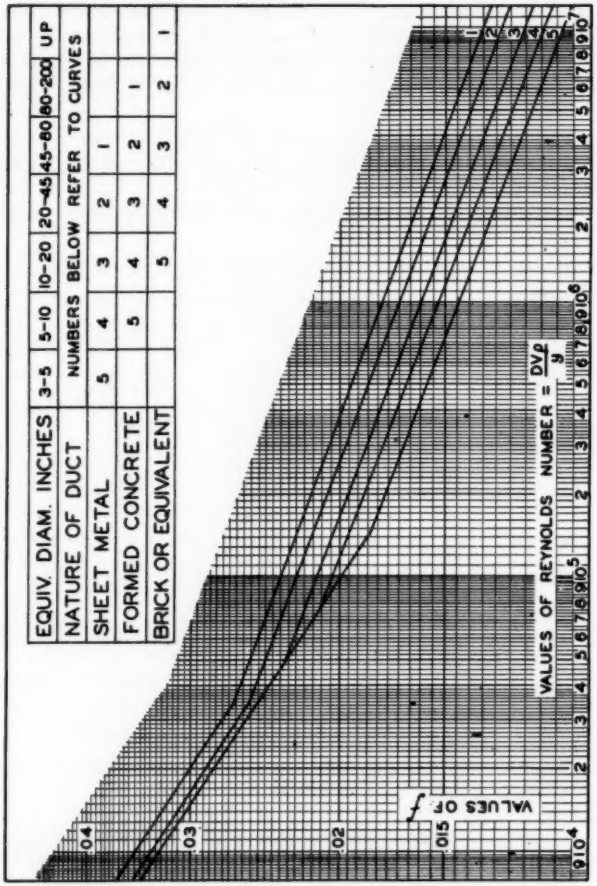

FIG. 5. VARIATION OF FRICTION FACTOR WITH REYNOLDS NUMBER

an equivalent length of 56.6 ft. Velocities may be roughly determined by the simple expedient of equating $\left(\frac{V_2}{V_1}\right)^2 = \frac{28.6}{56.6}$ whence $V_2 = 0.7\ V_1$.

The stack velocity will be $0.7 \times 750 = 525$ fpm and in branch duct $0.7 \times 1000 = 700$ fpm. Using the nearest commercial size the riser becomes 3 x 10 in. with an actual velocity of 480 fpm and the branch duct 3 x 7 in. at 687 fpm.

A summary of the actual design head losses to grille No. 4 is given in Table 4.

The difference between 0.0758 (Table 2) and 0.0485 (Table 4) or 0.0273 in.

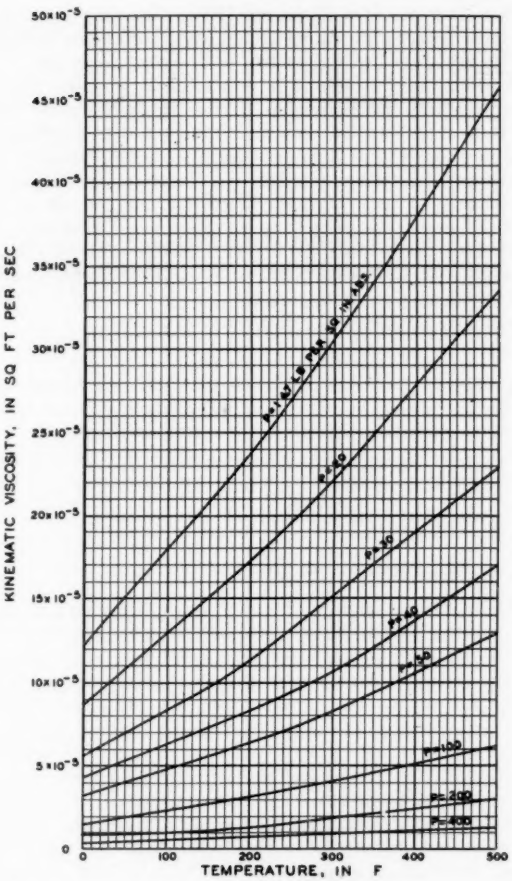

FIG. 6. CURVES FOR DETERMINING KINEMATIC VISCOSITY

w.g. is for 12:5 lineal feet carrying 705 cfm. By trial and error a 7 x 12 in. duct carrying the required quantity at an actual velocity of 1210 fpm indicates a drop of 0.025 in. which balances the pressures with accuracy equivalent to that of the usual heat loss calculations.

CONCLUSION

It is obvious that the method employed in this discussion is a combination of the *Velocity Method* and the *Friction Pressure Loss Method* referred to

on page 369 of THE A.S.H.V.E. GUIDE 1937. The only velocities arbitrarily assigned are those where high values would lead to noise, or turbulence, or duct vibration. The remainder of the system is designed for equal overall friction drop without attempting to adhere to any specific pressure loss per foot of length. It is purposely left somewhat flexible in order to use, wherever possible, stock commercial sizes of duct and fittings and to permit other freedom of design. The ultimate aim is to be able to so design the system that it will reduce the amount of artificial friction, *viz.* dampers, splitters, etc. to a minimum if not entirely to zero. It is believed that the cost of power consumed by such artificial resistance will in most cases more than offset the cost of investment in more ample duct sizes. This is the *reasoning* involved in the method proposed. It is hoped that results will be commensurate with the slightly more involved plan of approach.

No. 1051

THE NOISE CHARACTERISTICS OF AIR SUPPLY OUTLETS

By D. J. STEWART * (*MEMBER*) AND G. F. DRAKE ** (*NON-MEMBER*), ROCKFORD, ILL.

NOISE is one of the first considerations which arises when one attempts to specify a suitable grille for an air supply outlet in a ventilating system. This paper summarizes the results of studies of the noise level of grilles of various types and sizes and with different air velocities. The data are presented in a form that can be used conveniently in a straightforward method of grille selection which is also described.

INFLUENCE OF NOISE ON GRILLE SELECTION

After the heating or cooling load calculations have been completed, a supply air temperature has been selected, and the volume of air required determined, the next step is to determine the velocity at which air may be introduced into the space without objectionable noise or objectionable drafts. In this discussion, the noise from the fan and duct system itself will be neglected and only the noise produced by the grille considered. If the fan and duct system is properly constructed and acoustically treated, the noise from this source can be neglected except at extremely low grille velocities. The noise level of the open duct in the tests under consideration was below 23 decibels (db) at an outlet velocity of 1000 fpm. Since high velocities mean smaller ducts and outlets, it is advantageous economically to use the maximum velocity permitted by the desired noise level or by the distribution characteristics. The selection of the proper velocity requires that the designer have available reliable data on the noise characteristics, applicable to the particular make of outlet it is proposed to use. Before a final selection can be made, the designer must also have data on the distribution characteristics, but that phase of the subject will not be discussed in this paper.

METER LOCATION

It is assumed that the reader is familiar with the elementary principles of the theory of sound as well as of the decibel scale. All intensity figures given

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in this paper are on the decibel scale and refer to the now generally accepted threshold, 10^{-16} watts (w) per square centimeter.

When air is introduced into a room through a grille, sound energy is produced at the grille; and if the air is at a constant velocity, the sound energy

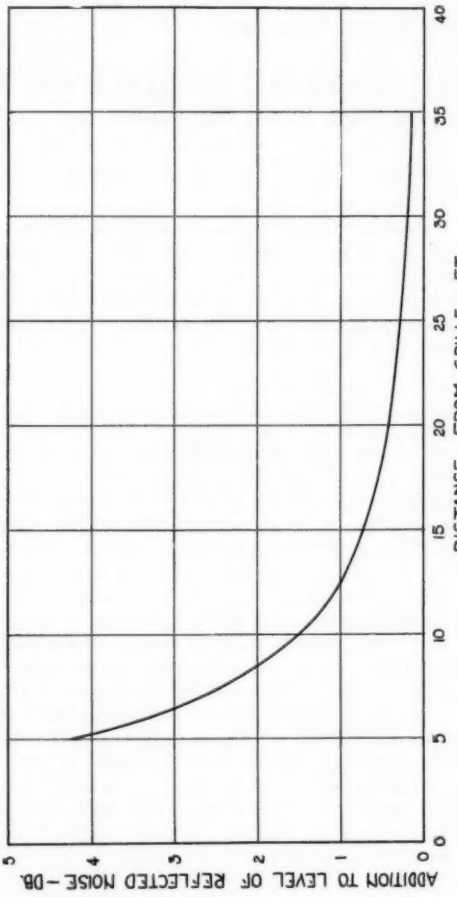

FIG. 1. DECIBEL ADDITION TO LOUDNESS LEVEL OF REFLECTED SOUND. ROOM ABSORPTION 270 SABINS. LONG THROW GRILLE

is, for all practical purposes, produced at a constant rate. If there were no reflections, the intensity of the sound would be approximately proportional to the rate at which the energy is generated and inversely to the square of the distance between the grille and the observer. This is not the case in an ordinary

room, for due to partial reflection at the boundaries the intensity of sound at any point in the space builds up immediately to a constant value higher than that which would be indicated by the rule given. In a large room at a point remote from the source of sound, the intensity can be shown to be substantially proportional to the rate at which sound is generated and inversely proportional to the number of sound absorption units in the room and independent of the distance from the outlet. It is not satisfactory to consider the grille noise on this latter basis, since the occupants of the room may be very close to the grille.

The authors know of no generally accepted standards as to the place at which sound measurements should be made, but it appears unlikely that an occupant will be nearer than directly in front of the grille, 5 ft from the lower edge of the outlet, this distance measured downward at an angle of 45 deg. All measurements referred to in this paper were made at this position. Fig. 1 indicates the manner in which loudness level varies with position.

ROOM ABSORPTION

The effect of the absorption characteristics of the room on the intensity at this point is not nearly so great as at remote points in the room, but nevertheless cannot be ignored completely.

It can be shown that the ratio between direct and reflected sound intensity in a room is expressible as:

$$R = \frac{SC}{16\pi d^2} \times \frac{\alpha}{1-\alpha} \qquad [1]$$

where

S = surface area of the room,
d = distance from the source,
α = average absorption coefficient,
C = directivity of the sound source.

For a given type of grille the value of C can be determined from this equation as follows: the distance d is that from the source to the standard listening position (5 ft). S can be calculated from test room dimensions, and α determined from published data on the absorption coefficients of various construction materials. If now the intensities at distance d and at some point remote from the source (so that direct sound is negligible) are measured, the ratio R may be computed as:

$$R = \frac{I_T - I_R}{I_R} \qquad [2]$$

where

I_T = total intensity as measured at d,
I_R = reflected intensity as measured at the remote point, the difference being the direct intensity at d.

All the unknowns of equation (1) except C are now established, and C can therefore be computed. By this semi-empirical means it is possible to establish a relation which, by justifiable approximations, can be simplified to:

$$\Delta L = 10 \log \left(1 + \frac{K}{a}\right) \qquad [3]$$

where

ΔL = loudness addition in decibels,
K = constant including the factor C,
a = absorption of the room in *sabins*.

From this expression it may be seen that when the absorption is infinite (as in free space) the addition ΔL is zero, as would be expected. From this

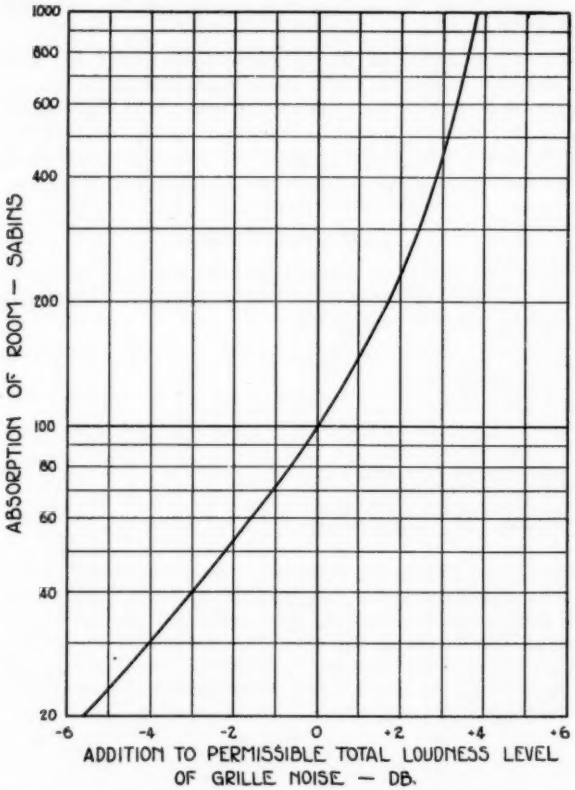

FIG. 2. DECIBEL CORRECTION FOR ROOM ABSORPTION. REFERRED TO ROOM OF 100 SABINS

equation a curve could be plotted showing the additions in decibels which should be made to the *free space loudness* of a particular type grille when placed in a room of absorption a. In order that such a curve might be more directly applicable, a base of 100 sabins was chosen and the corrections referred to a room having this absorption. Although the curve shown in Fig. 2 is for only one type of grille, it may be used for any type of grille with suffi-

cient accuracy for all practical purposes. The sound absorption of the test room is approximately 270 sabins, but all the data shown herein are corrected to an absorption of 100 sabins. An average room will have an absorption of more than 100 sabins, and failure to correct for this factor will, therefore, give a permissible sound level lower than that which actually could be used. The error in this case is on the side of safety. It is only where the room is highly reflecting or very small that there is any great necessity of taking this factor into account. The total absorption coefficient can be computed from the usual acoustical formula. A table of the coefficients of most of the usual interior finishes is given on page 331 of THE A.S.H.V.E. GUIDE, 1936.

LOUDNESS VS. VELOCITY

In Fig. 3 are shown as circles a number of test values of loudness level for a long throw type grille of one square foot area (corrected to a room of 100 sabins absorption) with various face velocities. A curve such as the full line of Fig. 3 might be considered a fair representation of these data. Consideration of this curve and the physics of the problem suggested that the sound intensity might be proportional to some power of the face velocity. This led to the derivation of the following equation:

$$L_s = b \log V + c \qquad [4]$$

where

L_s = loudness for 1 sq ft core area (specific loudness),
V = face velocity (feet per minute),
b and c = constants dependent upon grille.

The determination of constants b and c by substitution of two sets of test data in the above equation gave the following results for the long throw type grille:

$$c = -91.1$$
$$b = \ \ 44.0$$

Substitution of these values in (4) gives:

$$L_s = 44 \log V - 91.1 \qquad [5]$$

The full and dotted line of Fig. 3 is the graphical representation of (5). This curve may be seen to check almost exactly at all measured values of L_s, passing, of course, through the two points used in determining b and c. Below these points (dotted line) no satisfactory check has yet been made, because of the low general noise level required, but the shape of the curve seems reasonable. It should be noted that the curve reaches zero loudness level at about 118 fpm face velocity. This does not mean that no sound energy is produced at this velocity, but that it is at the threshold of audibility in a room of absolute silence. If the room noise level is itself at the threshold, the combination would produce a noise of 3 db since it would be twice threshold intensity. Further examination of this curve and consideration of its equation shows that it approaches the zero velocity line asymptotically, reaching it at minus infinity loudness level;

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this means only that the sound energy, the logarithm of which (in threshold units) is the loudness level, reaches zero at zero velocity, as might be expected. It is intended to check this curve experimentally at somewhat lower velocities when a better location and better apparatus are available, but it is believed safe for practical purposes.

CORRECTION FOR GRILLE AREA

Correction must be made for area to obtain the total sound level. This can be done by the use of the formula:

$$D \text{ (decibel addition for area)} = 10 \log_{10} A \qquad [6]$$

where

A = core area in square feet.

Since the allowable total sound and the required volume of air are usually known, and it is desired to determine the maximum allowable velocity, the chart and formula given involve a trial and error method, for the volume cannot be determined until an assumed velocity has been decided upon; and then the grille size, unless it happens to be one square foot core area, will indicate a different sound level than that originally chosen. To avoid this, a family of curves shown in Fig. 4 may be used. These curves are simply the relation between volume of air and face velocity at constant loudness. If the volume and desired total noise are known, the maximum permissible velocity can be directly determined. Fig. 4 is based on the same grille construction as Fig. 3. It is interesting to note that the curves in Fig. 4 are straight lines. The reason for this is given in the following discussion.

It has been shown that the loudness per square foot and the addition for area may be represented respectively by:

$$L_s = b \log V + c \qquad [4]$$

$$D = 10 \log A \qquad [6]$$

The value of A may be expressed in terms of velocity and rate of flow as:

$$A = \frac{Q}{V} \qquad [7]$$

where

Q = rate of flow in cubic feet per minute.

The total loudness level of a grille is:

$$L_T = L_s + D \qquad [8]$$

$$= b \log V + c + 10 \log \frac{Q}{V} \qquad [9]$$

Solving this for log Q,

$$\log Q = \frac{L_T}{10} - \frac{(b-10)}{10} \log V - \frac{c}{10} \qquad [10]$$

The constants b and c are the same. Since both Q and V appear as logarithms, the relation between them will appear as a straight line on logarithmic

paper. Various values for L_T determine, therefore, a family of straight lines on logarithmic paper.

PERMISSIBLE SOUND LEVEL

In order to determine the permissible sound level, it is necessary either to know or assume the noise level of the room without the grille noise. In many

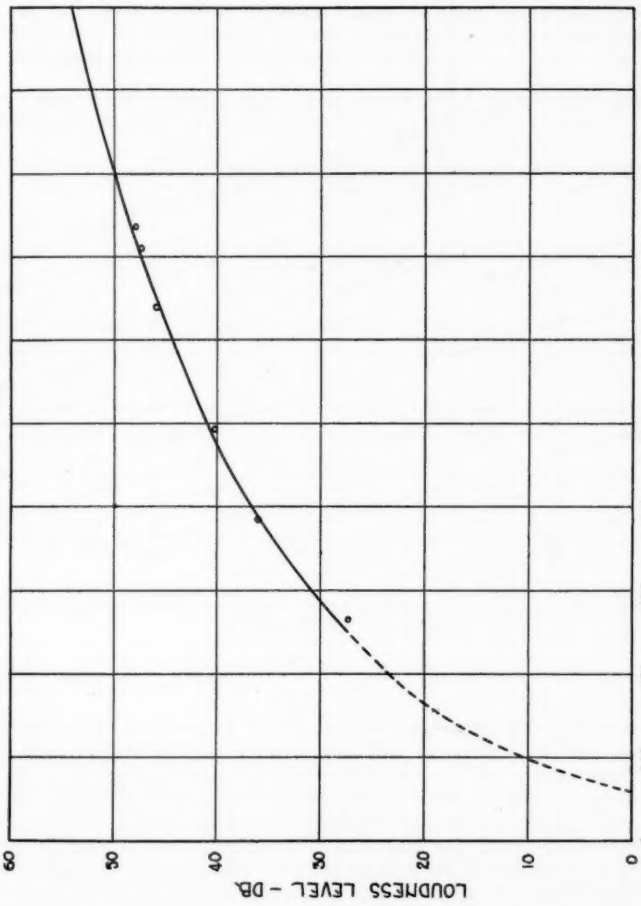

FIG. 3. SPECIFIC LOUDNESS OF LONG THROW GRILLE. ROOM ABSORPTION 100 SABINS

rooms the noise level will vary, and it is usually necessary to design the system for satisfactory performance with the minimum noise level which it will have when occupied. If the noise level without the air conditioning system is known and the permissible noise level with the system in operation is known, the permissible maximum noise level due to the grille itself may be computed. The

alignment chart, Fig. 5, solves this problem directly and eliminates the necessity of any computation.

Experience indicates that an increase in noise level of 1 db (at about 30 db) can hardly be detected, but that 2 db increase can be detected. If the fan is started and stopped during occupancy, an increase of more than 1 db is usually objectionable; and by referring to the chart it is found that this requires that the noise level of the grille be about 5 db below the noise level of the room without the grille. On the other hand, if the fan runs continuously, an addition of 3 db or even more is often entirely unobjectionable. An increase of 3 db in

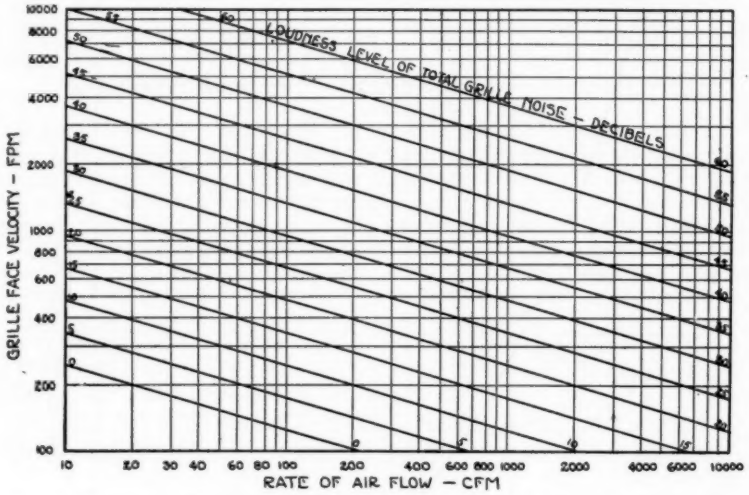

FIG. 4. FAMILY OF CONSTANT LOUDNESS LINES FOR LONG THROW GRILLE. ROOM ABSORPTION 100 SABINS

total noise permits the grille noise alone to be equal to the room noise without the grille.

When more than one outlet must be considered, the problem of whether to consider the total air volume or that from one outlet is dependent upon the location of the outlets and the sound absorption characteristics of the room. If two outlets are located some distance apart in a highly absorbent room, the noise level can be computed on the basis of a single outlet. On the other hand, if the room is small, and especially if it is highly reverberant, the noise level should be computed on the basis of the total volume of all the outlets. As the latter assumption errs in the direction of safety, it is the usual practice.

EXAMPLE

Assume that the load calculations have been made and the supply air temperature selected, and that as a result the volume of supply air has been determined to be 900 cfm.

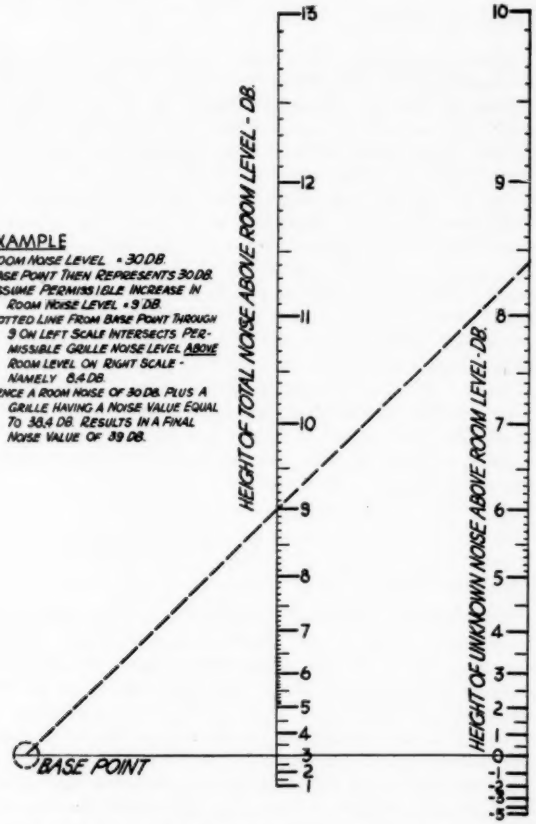
EXAMPLE
ROOM NOISE LEVEL = 30 DB.
BASE POINT THEN REPRESENTS 30 DB.
ASSUME PERMISSIBLE INCREASE IN ROOM NOISE LEVEL = 9 DB.
DOTTED LINE FROM BASE POINT THROUGH 9 ON LEFT SCALE INTERSECTS PERMISSIBLE GRILLE NOISE LEVEL ABOVE ROOM LEVEL ON RIGHT SCALE - NAMELY 8.4 DB.
HENCE A ROOM NOISE OF 30 DB. PLUS A GRILLE HAVING A NOISE VALUE EQUAL TO 38.4 DB. RESULTS IN A FINAL NOISE VALUE OF 39 DB.
HEIGHT OF TOTAL NOISE ABOVE ROOM LEVEL - DB.
HEIGHT OF UNKNOWN NOISE ABOVE ROOM LEVEL - DB.
BASE POINT

FIG. 5. CHART FOR DETERMINING INCREASE IN ROOM NOISE LEVEL DUE TO ADDITION OF GRILLE NOISE

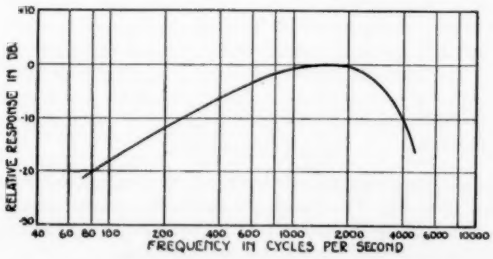

FIG. 6. APPROXIMATE OVERALL FREQUENCY WEIGHTING CHARACTERISTIC OF NOISE METER

Design Conditions

The room in which this system is to be installed is quite reverberant and has an absorption of only 50 sabins. The room noise level without the air conditioning system has been determined to be 45 db, and the air conditioning system itself has been quieted so that the noise from it can be neglected. On account of the small volume of air, it is probable that only one supply outlet will be used in this instance, but even if more than one supply outlet is contemplated it will be wise to consider the total volume of air as concentrated at one point since the absorption coefficient is so low. The system will not run continuously, and it is required that the additional noise of the grille will not be noticeable. As previously stated, an increase in the total noise level of approximately 1 db is not noticeable, and reference to Fig. 5 indicates that the grille noise level must be about 5 db below the room level or 40 db. However, the absorption coefficient of this room is only 50 sabins, and reference to Fig. 2 indicates that the permissible level of grille noise must be reduced 2 db on this account. The net permissible grille noise is thus 38 db, and by referring to Fig. 5 it will be seen that this corresponds to an air velocity of 880 fpm. It is possible that consideration of the flow and distribution characteristics may require a lower velocity, but no higher velocity may be used and still fulfill the requirements as to noise.

THE TEST PROCEDURE

The Test Room

The room in which the sound data described herein were taken is one which is regularly used for air distribution tests. The building itself is of poured concrete, pillar and slab construction, with a flat concrete ceiling covered with built-up composition roofing. The test room is on the sixth floor of this building, and is 37 ft long, 17 ft wide, and 9 ft 7 in. high. The long side walls are of sheet metal, outside of which are narrow spaces adapted to be heated to maintain room temperature during tests with cool air. The ceiling is ¼ in. rigid insulation on joists about a foot below the concrete slab of the roof. The floor is concrete. The grilles are located for test in the center of one end wall and about 7 ft 6 in. from the floor. An exhaust grille, extra large to reduce velocity and therefore noise to a negligible value, is located in the center of the same wall at the floor. This wall is also of ¼ in. rigid insulation and forms the front of a pressure box 5 ft deep and 8 ft wide extending from the top of the exhaust grille to the ceiling. Air is introduced into this box near the bottom and at velocities greatly reduced by a diverging nozzle with splitters. The duct supplying this nozzle is acoustically treated to eliminate noise from the fan system. The exhaust or return duct is similarly treated. Behind the test grille is mounted a 3-ft section of duct to simulate an actual installation. Careful tests have indicated that the flow of air from the pressure box through this duct is uniform over its entire cross-section.

Although the metal walls and concrete floor are highly reflecting, the unpitched character of the noise (see Fig. 7) offered no difficulties due to standing waves, and the ceiling and end wall material brought the absorption of the room as a whole to approximately 270 sabins (by computation).

The Fan System

The source of air was the fan of an air conditioning system used for flow and distribution tests on grilles. This is a standard direct expansion refrigerant installation of 10 ton capacity with provision for reheating and humidifying. Two multi-speed direct current motors are available to give fan speeds of 25 to 700 rpm. In the supply duct is a carefully calibrated Venturi nozzle having an 18 in. approach duct and 10 in. diameter throat. Pressure lines are carried from this nozzle to a small room off the test room where the pressures are measured accurately with a Wahlen gage capable of detecting a pressure difference of one-half thousandth of an inch of water. By these means the volume of air being supplied to the pressure box is measured rapidly and with considerable accuracy. A slope gage is provided for measuring the static pressure in the box and a curve of pressure versus cubic feet per minute leakage with grille opening closed provides means of making correction for this source of error which is, however, quite small. A remotely operable butterfly damper in the supply was used to secure final adjustment of air flow.

The Sound Level Meter

All sound measurements were made with a standard type sound meter graduated in decibels, but not above the now accepted standard of 10^{-16} w per square centimeter. In addition, a separate amplifier having a gain of about 25 db was used on the low velocity measurements. All data taken were corrected to the *A.S.A.* standard of 10^{-16} w per square centimeter by the addition of 7 db to the meter readings.

The frequency response characteristic of the sound level meter is approximately as shown in Fig. 6. This deviation from uniform response is accomplished by a frequency weighting network and is intended to approximate the response of the human ear at a loudness level of about 50 db. That this is not correct for other energy levels is clear by inspection of the equal loudness contours which are a part of the *American Tentative Standards for Noise Measurement* Z24.2, 1936. However, because the frequency of the most prominent grille noise components appears to lie between 500 and 2000 cycles (see Fig. 7), the error introduced at loudness levels other than 50 db is less important than it might otherwise be. It is proposed that these data be checked with a sound level meter, when available, which meets the *American Tentative Standards for Sound Level Meters* Z24.3, 1936.

The Test Data

As stated before, the standard listening position was chosen at a distance of 5 ft from the lower edge of the grille downward at 45 deg in a plane perpendicular to the grille face at its center. A stand was provided for holding the noise meter with its microphone facing the grille at this position. With the fan running, the butterfly damper was closed to stop air flow, and the room noise level measured with the noise meter in standard position. The air flow was then adjusted to approximately the desired value, the actual flow through

the Venturi nozzle measured, the static pressure in the box noted and, from the leakage curve, correction made to give net flow through the grille. The net core area having been carefully measured, a simple division gives the average face velocity. A noise level reading with this air flow determines one point on the curve of noise level versus velocity for this particular grille type and area.

The corrections previously described permit conversion to loudness of the grille noise only per square foot of core area in a room of 100 sabins absorption.

A number of readings were taken at other than the standard listening position in order to determine the relation of sound level at these positions to that of the standard position. The curve of Fig. 1 represents quite closely the locus of these test data. The level of the purely reflected sound is substantially that at the most remote point of the room. The curve of Fig. 1 was determined mathematically in the following manner.

After c has been established, equation (1) may be reduced to:

$$R = \frac{K}{d^2} \qquad [11]$$

and from equations (11) and (2)

$$I_T - I_R = \frac{K}{d^2} I_R \qquad [12]$$

or

$$I_T = \left(\frac{K}{d^2} + 1\right) I_R \qquad [13]$$

The difference in decibels between the level of purely reflected sound and the total sound may be expressed as the logarithm of their intensity ratio:

$$L_T - L_R = 10 \log \frac{I_T}{I_R} \qquad [14]$$

From equations (13) and (14)

$$L_T - L_R = 10 \log \left(\frac{K}{d^2} + 1\right) \qquad [15]$$

Sources of Error

There are several obvious sources of error. First, the measurement of air flow depends upon the accuracy of calibration of the Venturi; at the rates of flow used, this calibration is probably accurate to ± 1 per cent. The error in reading is approximately of the same order, for although the sensitivity of the Wahlen gage is much better than this, the fluctuations in pressure add appreciably to the error in reading. Although the determination of leakage at a given pressure may be in error by ± 10 per cent, this leakage is such a small part of the total flow that the error in net flow due to this is probably less than ± 1 per cent. Errors in measurement of core area are negligible

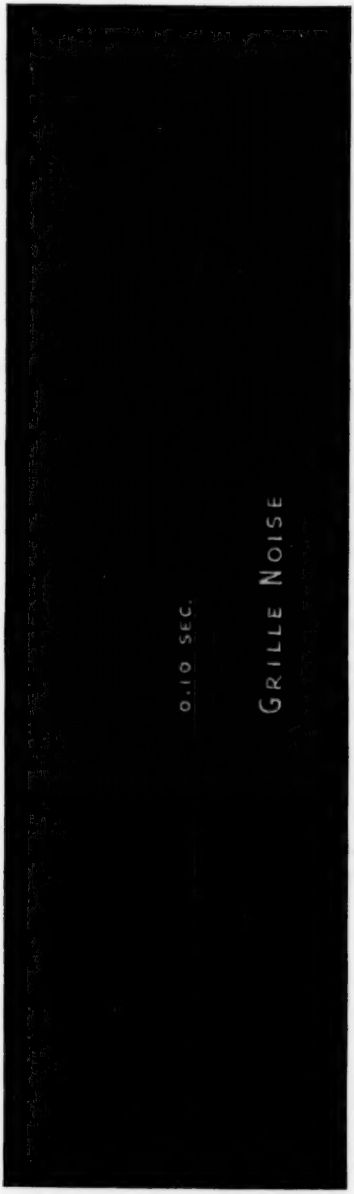

FIG. 7. OSCILLOGRAM SHOWING FREQUENCY CHARACTERISTICS OF GRILLE NOISE

relative to the above. Thus, it would appear that the maximum error in average face velocity as recorded is ± 3 per cent. Because of the way in which the loudness level varies with velocity (see equation (5)), this is equivalent to less than ± 0.6 db.

Means to check the absolute values of the sound meter readings were not available. However, the instrument was calibrated by the manufacturer, and it was possible to check from time to time the gain of the amplifying system. It is not likely that the dynamic system of the microphone suffered any considerable change during the testing period, so that even if the absolute value is in error the relative values are probably quite accurate. The meter itself has a sufficiently open scale so that it is easily read to 0.5 db, although at the very low levels the fluctuation of extraneous noise increases this error to about 1 db. The gain of the external amplifier was checked frequently by taking the meter readings with and without the amplifier for a constant sound high enough in intensity to be readable without the amplifier.

It appears probable that, assuming no error in absolute value of loudness level, the total of all remaining errors did not exceed ± 1 db for a single observation, except perhaps at the very lowest velocities. Inasmuch as several observations were

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made for each point, the probable error is believed to be approximately ± 0.5 db, again excepting the error in absolute loudness level.

DISCUSSION

H. M. HART: I would like to know what type of grille was used, and if the type has anything to do with the sound level.

S. H. DOWNS: I would like to ask if the height of the grille above the floor has any effect on the noise, or if the grating is horizontal or vertical.

J. H. VAN ALSBURG: Sound, to many of us appears to be a very complicated bit of engineering work, but in the form given here, I believe, is very simple though it requires a little study. I think it would be well for many of us to spend a little time thinking about it and we will find that it is applicable to about all the work before us.

I have just one question in connection with this paper. In all the sound laboratories that I have visited, where research has been undertaken, because of the variation in centers of reflected noise, it has usually been advisable to install a rotating piece of metal to equalize the reflection of noise. Has this been attempted in connection with these tests?

D. J. STEWART: One family of curves shown was for a fin-type grille with horizontal fins. There is some difference, not in the shape of the curves but in the values, depending upon the type of grille used. Even vertical fins as opposed to horizontal fins makes a considerable difference in the volume of noise.

The answer to the second question, of Mr. Downs, is: It does. The assumption of 5 ft from the grille at an angle of 45 deg is wholly arbitrary. We don't know if that is correct or not, but that is about as near as the observer is ever apt to be located. In a very high room, for example, it may well be 10 ft from the grille to the nearest possible observer, in which case the noise of the grill will be reduced by 2½ decibels.

Mr. Drake will answer Mr. Van Alsburg's question.

G. F. DRAKE: The rotating blade or equivalent to prevent standing waves in the room was found by experiment to be unnecessary in this case because of the unpitched character of the noise. If you will notice the oscillogram accompanying this paper, you will see that the frequency of the noise generated by the grille is more or less random. Of course, there is a general characteristic frequency somewhere around 2000, and another around 500, but they are not steady enough to give us any trouble due to standing waves in the room, as we determined by measurement, moving a microphone around the room.

No. 1052

THE NATURE OF NOISE IN VENTILATING SYSTEMS AND METHODS FOR ITS ELIMINATION

By J. S. PARKINSON * (*NON-MEMBER*), MANVILLE, N. J.

NOISE control is becoming increasingly important in the design of ventilating and air conditioning systems. This is partly because conditioning is being used in office and residential locations where the requirements for noise control are more rigid, and partly because of an increasing noise consciousness on the part of the buying public. There is, however, little accurate engineering information available. The following discussion presents certain definite facts which may be used as a foundation and indicates where further work must be done. It is presented only as a progress report.

In cooperation with the A.S.H.V.E. Technical Advisory Committee on Sound in Relation to Heating and Ventilation, the author has prepared a table of typical noise levels as they are encountered in various room locations. These values are presented in Table 1. It has been suggested that as a temporary procedure the equipment noise level should not be greater than the background noise level already existing in the space. This means that when the equipment noise is present the total noise will be increased 3 db (decibels). In cases where the equipment noise must be practically eliminated, it will be necessary to reduce such noise to a point about 10 db below the background level. Such a procedure disregards the effect of frequency variations, but since the problem of annoyance versus frequency distribution is so complex, no simple specification can be set up to cover this point. In general, though, it may be said that the loudness level at any frequency should not be greater than the background noise, and at the high frequencies the loudness level of the equipment noise should if possible be materially lower.

Two curves (due to Hale Sabine) are shown in Fig. 1 which may be called *equal annoyance contours.* Since there are no exact data on the annoying or disturbing effect of various frequencies, such a curve can only be an estimate. The general form of these curves is, however, substantiated by numerous tests. The curves are drawn so as to pass though a loudness level for each frequency which is equally as annoying as any other point on the curve. Thus it may be estimated that a loudness level of 40 db at 5000 cycles is just as annoying as one of 55 db at 1000 cycles, and that at 100 cycles it requires a loudness

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level of 68 db to be as annoying. The important fact to observe is that annoyance increases with frequency, and levels which are not disturbing at low frequencies may in consequence become troublesome at high frequencies.

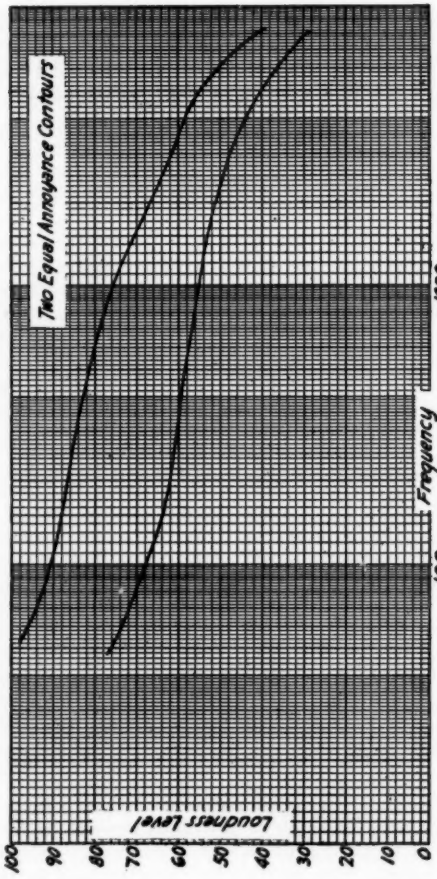

FIG. 1. RELATION BETWEEN FREQUENCY AND DEGREE OF ANNOYANCE FOR LOUDNESS LEVELS OF 35 DB AND 75 DB (ESTIMATED)

CHARACTERISTICS OF NOISE IN VENTILATING SYSTEMS

In order to meet these requirements it is necessary to know the nature and source of the noise caused by ventilating systems. It must be determined where these sounds originated, how they are transmitted, and what measures can be taken to eliminate them. It is not the purpose of this paper to discuss

TABLE 1—TYPICAL NOISE LEVELS IN VARIOUS LOCATIONS

ROOM LOCATIONS	LOUDNESS LEVEL IN DECIBELS TO BE ANTICIPATED[a]		
	Min.	Representative	Max.
Sound Film Studios	10	14	20
Radio Broadcasting Studios	10	14	20
Planetarium	15	20	25
Residence Apartments, etc.	25	35	40
Theaters—Legitimate	25	30	35
Theaters—Motion Picture	30	35	40
Auditoriums, Concert Halls, etc.	25	30	40
Churches	25	30	35
Executive Offices—Treated Private Offices	25	33	40
Private Offices—Untreated	35	45	50
General Offices	45	55	60
Hospitals	25	40	55
Class Rooms	30	35	45
Libraries, Museums, Art Galleries	30	40	45
Public Building—Court Houses, Post Offices, etc.	45	55	60
Small Stores	40	50	60
Upper Floors Department Stores	40	50	55
Stores—General—Including Main Floor Dept. Stores	50	60	70
Hotel Dining Rooms	40	50	60
Restaurants and Cafeterias	50	60	70
Banking Rooms	50	55	60
Factories	60	70	80
Office Machine Rooms	60	70	80
VEHICULAR NOISE			
Railroad Coach	60[b]	70	80
Pullman Car	55[b]	65	75
Automobile	50	65	80
Vehicular Tunnel	75	85	95
Airplane	80	85	100

[a] It is assumed that if equipment noise alone is not greater than these levels of background noise at any frequency band, results will be commercially satisfactory. When both are heard together, an increase in total level of 3 db can be anticipated.

[b] For train standing in station a level of about 45 db is the maximum which can ordinarily be tolerated.

TABLE 2—LOUDNESS LEVEL IN DECIBELS AT VARIOUS SPEED RANGES

FREQUENCY	700–780 FPM	1234–1570 FPM	1760–2085 FPM	2580–3440 FPM
Total	59	77	59	74
0–64	5	20	14	35
64–128	19	41	24	45
128–256	58	72	58	69
256–512	53	75	53	73
512–1024	47	66	45	61
1024–2048	45	65	43	58
2048–4096	42	63	42	58

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the mechanical design of equipment in other than general terms, since this is a problem which the manufacturer is better equipped to handle than the acoustical engineer.

The noise encountered in ventilating systems has three general sources, the motor, the fan, and the air itself. The motor generates frequencies which are related to the rotational speed, and the electrical frequency. The fan may generate sounds which are a function of its rotational frequency, and will also generate frequencies which are multiples of the blade frequency. The air moving past roughnesses or obstructions in the duct is set into vortices which give off high frequency sound. There may also be noise generated at the outlet which results from the abrupt change in dimensions or from faulty grille design.

Usually of particular interest is the noise at a point a few feet from the outlet of the duct where it enters the room. Measurements in various locations have shown that this loudness level may vary over a wide range—from 50 to 90 db above 10^{-16} watts/sq cm. A fairly representative range for untreated systems is from 55 to 75 db. In a small experimental system set up at the laboratory, consisting of a large blower (63 forward curved blades) connected to about 20 ft of 18 in. x 18 in. galvanized iron duct, the frequency analyses for different speeds gave the loudness levels shown in Table 2.

In order to change air speed the opening at the intake was constricted by a circular insulating board baffle and two different size pulleys were used on the fan. The baffle was used in making the measurements reported in the first two columns. The pulley diameter was 22 in. in column one and column three, 12 in. in column two and column four.

Two more ventilating systems were investigated under actual conditions of use. System A was an exhaust system connected to the blower through about 30 ft of 6 in. x 20 in. duct, opening into a small office through an opening in the side. The motor in this system was definitely noisy. System B was a pressure system opening into a small office through about 30 ft of 10 in. diameter round duct with a right angle bend at the opening and three more such bends between the opening and the fan. These systems were both sufficiently noisy to cause complaints. Air speeds were in the normal range for such installations. The frequency distributions were as follows, expressed in terms of loudness level:

	0–64	64–128	128–256	256–512	512–1024	1024–2048	2048–4096	TOTAL
System A.....	51	55	72	65	63	55	51	73
System B.....	62	57	69	66	55	43	36	71

In four cases out of these six the frequency band from 128 to 256 cycles contained the highest amount of energy. In the remaining cases the next highest frequency band—from 256 to 512 cycles—became more prominent as the air speed was increased. In every system measured during the course of these experiments the major part of the energy lay in this range. Therefore, if a reduction in the total loudness level is desired, a reduction in this range of

frequencies is the first essential. Since decibel levels must be added logarithmically to arrive at a total, even a large decibel reduction in a high frequency band will make only a small fraction of a decibel change in the total loudness level. On the other hand, a reduction in one of the higher bands may make a very appreciable difference in the annoyance factor. In general, it will make a difference whether a noise level guarantee is involved. Such guarantees are almost invariably based on the total noise level, and in such a case the low frequency range must receive first attention. If no guarantee has been made, and it is desired merely to make the noise less disturbing, a treatment to reduce

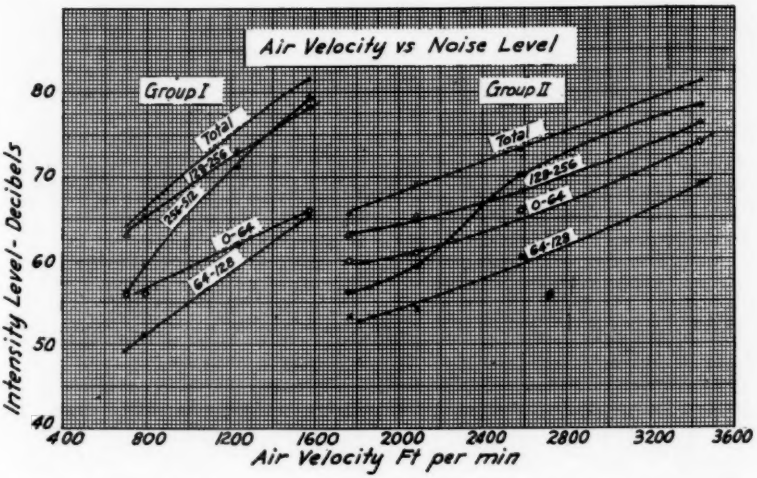

FIG. 2. INCREASE IN INTENSITY LEVEL WITH INCREASE IN AIR SPEED. THE RANGE OF FAN SPEEDS IS THE SAME IN BOTH GROUPS, BUT GROUP 1 USED AN INTAKE RESTRICTION WHERE GROUP 2 DID NOT

the high frequencies may be adequate. This will be discussed more in detail when the effect of sound absorbent materials is considered.

Table 2 shows certain useful facts about the relation between noise and air speed. When the air speed was increased by decreasing the pulley size and hence increasing the fan speed, the noise level increased. Compare columns 1 with 2, and 3 with 4. On the other hand, increasing air speed by increasing the size of the intake opening scarcely affected the noise level and actually decreased it in some cases. This can be seen by comparing columns 1 with 3, and 2 with 4. It is probable that the constriction placed over the intake interfered with the uniformity of air flow into the blades, thus increasing the noise level instead of decreasing it. These deductions corroborate a fact which has been observed elsewhere, that noise level is a function of fan speed rather than of air speed.

Additional data on the variation of noise level with air speed and fan speed are presented in Figs. 2 and 3. In this case the ordinates are intensity levels

in decibels without correction to the ear threshold since the variations of the level with air speed and fan speed were of sole interest. See also Table 3.

In this case the fan speed was the same in both groups. The increases were obtained first by increasing the motor speed and then by decreasing the pulley diameter. Here also the levels are related directly to fan speed but not to air speed. The frequency range may be divided readily into two groups. The frequencies below 512 cycles exhibit a fairly constant relation between fan speed and noise level with the exception of the 256-512 band in group one. This

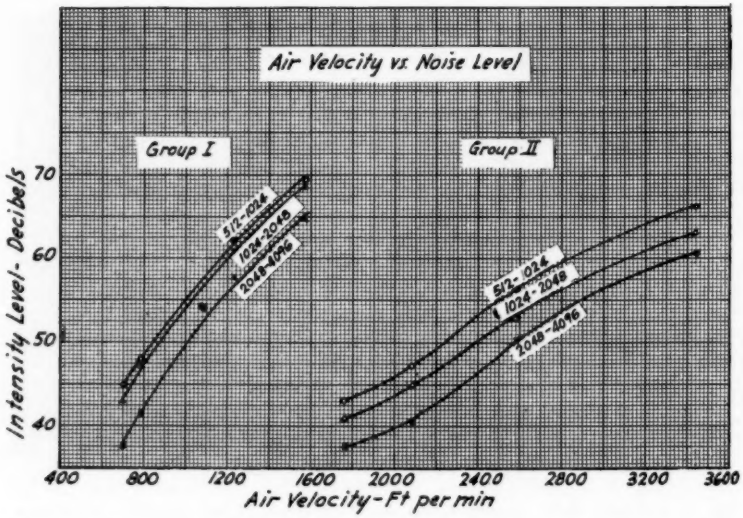

FIG. 3. INCREASE IN INTENSITY LEVEL WITH INCREASE IN AIR SPEED. THE RANGE OF FAN SPEEDS IS THE SAME IN BOTH GROUPS, BUT GROUP 1 USED AN INTAKE RESTRICTION WHERE GROUP 2 DID NOT

appears to belong more closely with the high frequency group. Those above 512 cycles also exhibit a fairly constant increase in noise level with fan speed, but the slopes of the curves are different from those in the low frequency group. This is interpreted to mean that the source of noise in the two frequency ranges is different. As suggested, it is probable that the upper frequencies are largely created by air turbulence.

METHODS OF ELIMINATING NOISE

There are a number of methods of preventing noise in ventilating systems, most of which are familiar. Vibrations can be prevented from telephoning through the walls of a duct by using suitable flexible connections between the fan and the duct. The fan and motor should be separated from the building structure by vibration insulating mounting. Fan speeds should be kept as low

as possible, and equipment should be well balanced and free from excessive vibration. Large duct sections should be braced to avoid resonance and diaphragmatic response. Construction should be such as to avoid unnecessary air turbulence. All these are precautions which any competent engineer will observe.

But there are many cases where even with all precautions the system is still unsatisfactory. In such cases sound absorbent lining placed inside the duct can be used with good effect. In choosing such a lining, a number of factors must be considered. The most obvious is sound absorbing efficiency, especially in the range below 500 cycles. Nearly all acoustical materials display high efficiency at the upper frequencies, but special design is necessary to obtain

TABLE 3—INCREASE IN INTENSITY LEVEL AT VARIOUS FREQUENCY BANDS EXPRESSED IN DECIBELS PER VELOCITY INCREASE OF 100 FPM

	WITH CONSTRICTION ON INTAKE	WITHOUT CONSTRICTION ON INTAKE
Total	2.0 db	0.92 db
0–64	1.15	0.84
64–128	1.85	0.95
128–256	1.85	0.78
256–512	2.64	1.31
512–1024	2.75	1.37
1024–2048	2.94	1.31
2048–4096	3.13	1.38

efficiency in the low frequency range. If a porous material is used, it must possess a fairly high flow resistance.

In addition to a good absorption-frequency characteristic, the material must be resistant to moisture and vermin, non-combustible, and must offer little frictional resistance to the passage of air across its surface.

THEORY OF SOUND TRANSMISSION THROUGH LINED DUCTS

A sound wave generated within a three dimensional closed space will undergo reflections at a rate equal to

$$n = \frac{vs}{4V} \tag{1}$$

where n = number of reflections per second
s = surface area of enclosed space
V = volume of space
v = velocity of sound

The energy after n reflections will be

$$E = E_0 (1 - \alpha)^n \tag{2}$$

where E_0 = the initial energy
α = the absorption coefficient of the reflecting surface.

If sound traveling down a duct is considered as suffering a similar series of reflections, the analogue to the quantity $4V/s$, which represents the mean free

path in a three dimensional space, is for the two dimensional case, $\frac{\pi a}{p}$, and

$$n = \frac{v\,p}{\pi\,a}$$

where p = the perimeter and
a = the cross sectional area.

The energy absorbed in unit time in a three dimensional space is inversely proportional to the mean free path. On this theory, in a duct the energy absorbed in unit time (or, equivalently, in unit length) will be proportional to p/a or inversely proportional to the hydraulic radius. If the energy equation is written in a form suitable to express reductions, substituting for n:

$$\frac{E}{E_0} = \left(1 - \alpha\right)^{\frac{v\,p}{\pi\,a}} \tag{3}$$

and therefore

$$\log \frac{E_0}{E_1} = -\frac{v\,p}{\pi\,a} \log\,(1 - \alpha) \tag{4}$$

or Energy reduction in Decibels

$$R = -\frac{K p}{a} \log\,(1 - \alpha), \text{ for unit length,} \tag{5}$$

where K is the experimental constant, as yet undetermined. Some experimental constant must be introduced to allow for the difference between the absorption coefficient as measured for random incidence in the reverberation chamber and the coefficient which must be used for waves traveling parallel to the surface of the material. This and the question of transverse flow are both involved.

In ducts whose cross sectional dimensions are of the same order or smaller than the wave length, the random reflection hitherto assumed is more likely to be replaced by a plane wave form traveling down the length of the duct. In such a case the effect of the lining would be to strip energy from the edges of the wave. There will then be a flow of energy transversely across the face of the wave depending on the pressure gradient produced by the absorption of energy at the edges.[1]

No attempt will be made here to deduce a complete mathematical theory for this case, but certain facts are apparent. The energy absorbed from the edges of a plane wave will be governed considerably by the ratio of the wave area in contact with the duct walls to the total area of the wave. This will be again expressed by p/a. The energy ratio will be again a direct function of the absorbing efficiency with the length as an exponential. The question remains, however, as to whether a change in the experimental constant will take care of variations in the transverse flow with different absorbers and different shapes, or whether some power of the quantity p/a must be used. This question will be discussed further as the course of the experiments are described.

EXPERIMENTAL DATA

Single Frequency Tones

These experiments employed a loud speaker actuated by a vacuum tube oscil-

[1] Some Data on a Room Designed for Free Field Measurements, by E. H. Bedell (*Journal of the Acoustical Society of America*, October, 1936).

lator as a sound source. A warble of ±20 cycles was applied to the test tone. The length of test duct was about 10 ft, terminating in a right angle bend at the point where the duct opened into the test room. Dimensions of the duct

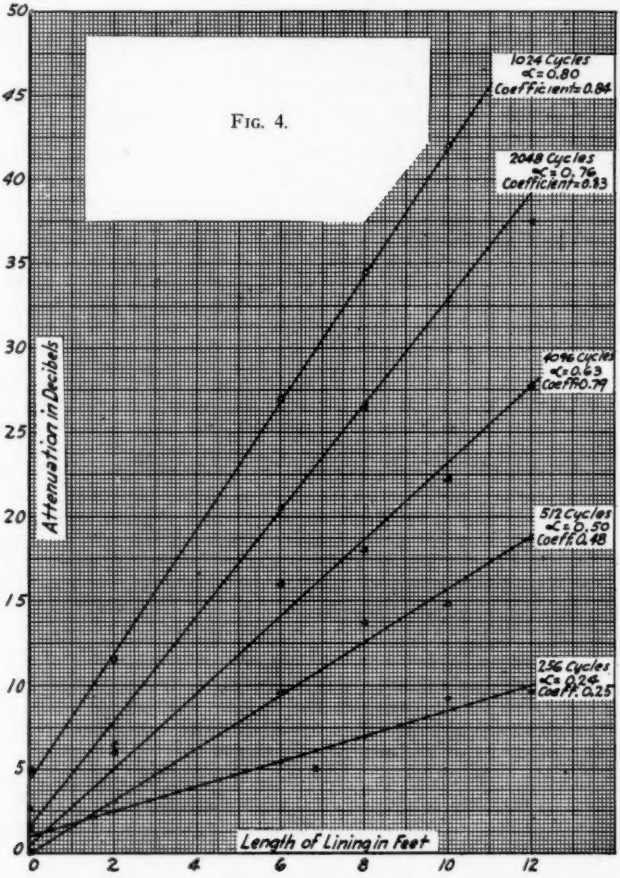

FIG. 4. ATTENUATION IN DECIBELS WITH INCREASING LENGTH OF ABSORBENT LINING

were 18 in. x 18 in. One microphone was placed at the beginning of the test length and a second at the outlet. Each was arranged in such a way that a number of positions could be taken to average out variations in the wave pattern. In making the experiments, succeeding sections 2 ft long were lined with absorbent material starting at the end nearest the source until the total

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length had been lined. In some cases the unlined duct itself showed an appreciable reduction.

A typical set of curves obtained from this test are illustrated in Fig. 4. The sound absorbent used had an absorption-frequency characteristic peaked at 1024 cycles. These curves demonstrated several useful facts. First, the assumption of a uniform decibel reduction per unit length is substantiated. Second, the reductions are uniform with length at all frequencies, and the slopes vary only with the absorption coefficient. This last fact was deduced as follows: Using the formula given under (5), K was found to be -1.95 for one frequency. As indicated in Fig. 4, the coefficients obtained at other frequencies from the slopes of the curve using this constant agree well with reverberation chamber coefficients. These experiments were repeated with a number of different types of linings, and the conclusions verified.

The next problem was to check the assumptions of the formula for different duct sizes. The above experiments dealt only with the single duct 18 in. x 18 in. Data were collected from other laboratories as well as for these experiments, and the results are shown in Table 4, column 3. Again all tests were made using single frequency tones, usually with a warble applied.

It was found that the original formula (5) gave results which were considerably too large for small duct sizes and large perimeters. Increasing the exponent of p does not correct this difficulty, for in any case where the exponent of p differs from the exponent of a an optimum duct shape will be found which is a better absorber than any other shape either larger or smaller, a result which appears untenable. The procedure finally used was to vary the exponent of the ratio (p/a) until a value was found which satisfied all conditions. Using this value a new experimental constant was computed. The best correlation to the data was found for exponents varying from 0.50 to 0.55, but since the experimental deviation was still considerably larger than the deviation produced by altering the exponent of (p/a) slightly, a value of 0.50 was finally selected. The formula then became:

$$R = -2.84\, l \left(\frac{p}{a}\right)^{\frac{1}{2}} \log\,(1-\alpha) \qquad (6)$$

(introducing the length factor l)

The data in column 4 of Table 4 are calculated from this formula (6) and the absorption coefficients of the materials as published.

While there are variations in many cases between measured and computed values there is no consistent trend in these variations, and the average error is less than 0.02 db. It therefore is possible that the variations observed are variations in absorbing efficiency or errors in measurement. For example, no account has been taken of the absorbing power of the duct itself when used in combination with a sound absorbent lining. There are many cases where such lining displays quite different properties when cemented to a sheet metal surface.

Elementary considerations would lead one to expect a constant much larger than 2.84, and it would be of interest to investigate this point. Since, however, the principal object of this work was to involve a simple working formula for engineering use, it has not as yet been possible to do this.

The formula in its present form indicates several facts of fundamental im-

portance. As long as the shape factor is constant, the larger the duct, the smaller will be the attenuation. The attenuation increases more and more rapidly with increasing values of α. It is therefore desirable to use absorbing materials having as high a coefficient as possible. For example, in any given duct a material having 75 per cent absorption will produce three times as much attenuation per lineal foot as one having a coefficient of 37½ per cent. Some of these conclusions are not new, but experimental verification has been lacking.

TABLE 4—DECIBEL REDUCTIONS FOR VARIOUS SIZE DUCTS

DUCT SIZE	FREQUENCY	MEASURED REDUCTION PER FOOT	CALCULATED REDUCTION PER FT FORMULA (6)
18 in. x 18 in.	256	0.62 db	0.34 db
18 in. x 18 in.	512	1.55	1.37
18 in. x 18 in.	1024	3.64	3.85
18 in. x 18 in.	2048	3.13	3.71
18 in. x 18 in.	4096	2.23	2.89
18 in. x 18 in.	256	0.44	0.31
18 in. x 18 in.	512	1.15	1.73
18 in. x 18 in.	1024	2.36	2.76
18 in. x 18 in.	2048	3.29	2.36
18 in. x 18 in.	4096	2.34	2.15
7⅞ in. x 17 in.	256	1.10	0.39
7⅞ in. x 17 in.	512	1.50	2.19
7⅞ in. x 17 in.	1024	3.65	3.48
7⅞ in. x 17 in.	2048	4.52	2.97
3⅜ in. x 17 in.	128	0.13	0.28
3⅜ in. x 17 in.	256	0.94	0.54
3⅜ in. x 17 in.	512	3.11	3.03
3⅜ in. x 17 in.	1024	4.20	4.81
3⅜ in. x 17 in.	2048	7.76	4.11
8 in. x 10 in.	1000	4.75	4.62
8 in. x 10 in.	1000	5.62	6.61
8 in. x 10 in.	512	1.88	1.98
6½ in. x 6½ in.	400	1.2	1.39
6½ in. x 6½ in.	1000	4.56	6.19
6 in. x 6 in.	512	2.56	2.92
6 in. x 6 in.	1024	4.00	4.64

Fan and Blower Noise

The final step in these experiments was to apply the conclusions obtained above to actual ventilating system noise. For this purpose a new test set-up was constructed with a large blower (63 forward curved blades, capable of delivering 6000 cfm at 200 rpm) in one room, a length of test duct leading through a carefully sound insulated wall into another room, and microphone at the outlet of the duct in the receiving room. The fan was of the multi-blade type, driven by a 5 hp electric motor. Both fan and motor were mounted on a vibration insulating platform to prevent direct communication through the floor. The fan was connected via a 39 in. adapter to the test duct, which was 18 in. x 18 in. and 14 ft long. The air was taken in and exhausted through large doors 3 ft 6 in. x 7 ft 0 in. The duct itself was wrapped with 2 in. of

rock wool blanket to avoid transmission through the walls into the receiving room. The microphone was placed in the test room at several positions within 30 in. of the end of the duct, but in no case beyond the plane of the end of the duct. Several tests were made to determine whether background noise or noise transmitted from the fan room by other means than through the duct could be interpreted in the results. In one case the duct was sealed off at a point before it entered the test room with an air tight seal. This procedure reduced the noise at all frequency bands to a point well below the levels measured during test. It appears safe to assume, therefore, that whatever sound reached the test room came through the duct. Transmission through the wall of the duct was prevented by a flexible canvas connection between fan and adapter section.

TABLE 5—REDUCTIONS IN NOISE LEVEL PRODUCED BY VARIOUS LENGTHS OF DUCT LINING

FREQUENCY BAND	REVERBERATION CHAMBER ABSORPTION COEFFICIENT OF MATERIAL	LARGE BLOWER—AIR VELOCITIES AVERAGED FROM 2600 TO 3700 FPM			
		2 Ft	4 Ft	6 Ft	8 Ft
0–64	...	2.1 db	4.0 db	6.2 db	...
64–128	0.08	1.5	1.0	3.2	3.2 db
128–256	0.33	2.2	1.9	3.9	3.5
256–512	0.56	3.2	5.0	6.6	7.7
512–1024	0.67	5.3	8.5	9.3	9.6
1024–2048	0.67	8.5	10.1	9.7	11.0
2048–4096	0.64	6.1	7.8	6.6	7.9
Total	...	1.5	2.6	3.1	4.0

Air velocity measurements were made with a Pitot tube and inclined water gage. Where necessary this gage was checked by a Wahlen gage. All air velocities given were the weighted average of seven Pitot positions taken across the duct. The Pitot tube was placed about five diameters from the outlet and six to seven diameters from the fan.

The noise analysis of this blower unit has already been given in Table 2 for various speeds. Typical reductions for the various frequency bands with succeeding lengths of lining material are given in Table 5.

It will be found upon analysis that the formula given holds fairly well for the first few feet, but beyond that point the reductions are less than expected. At the low frequencies this effect is less pronounced, but at the high frequencies the principal reduction occurs in the first few feet of lining.

In order to investigate this point the microphone was connected to the interior of the duct by a rubber tube at various points along its length. Starting at a point where there was only one foot of lining material between the tube position and the blower, successive positions were taken along the duct so that an increasing length of lining material intervened. The differences between

the base reading with 1 ft of lining effective and varying increments of length are given in Table 6. (Same material as in Table 5.)

At the high frequencies, the noise level appears to drop to a certain constant level after which further lengths of lining have little effect. To some extent this effect is observable at the low frequencies, but not to such a pronounced degree. The total noise reading, which is determined almost entirely by the low frequencies, displays this tendency only slightly. Since the experiments on single frequency tones all show a uniformly increasing decibel attenuation with increasing length, the explanation must be related to the conditions of this particular test. If the lining attenuates uniformly sounds originating at the fan, the only possible explanation is to assume another source of noise contributing to the level at the microphone. The experiments previously described indicate

TABLE 6—REDUCTIONS IN NOISE LEVEL PRODUCED BY ADDITIONAL LENGTHS OF DUCT LINING

FREQUENCY	4 FT ADDED	6 FT ADDED	8 FT ADDED	10 FT ADDED
0–64	4.5 db	5.7 db	7.7 db	7.0 db
64–128	5.7	7.3	9.7	8.3
128–256	3.1	4.3	5.2	5.5
256–512	3.1	4.3	6.9	6.5
512–1024	3.5	5.7	5.3	5.7
1024–2048	7.9	7.1	8.7	10.1
2048–4096	7.4	7.6	7.9	7.9
4096–8192	11.9	11.9	12.9	11.8
	4.4	5.4	6.4	7.7

that when the air is blocked off, there is little noise present. This process of elimination indicates that the air itself must be generating new sound as it passes through the duct.

If this is the case, it might be supposed that the air velocity would be a controlling factor, and that frictional effects at the surface of the lining material would be important. The early experiments were run at high air speeds. Accordingly experiments were tried over a lower range of air speeds from 700 fpm to 2100 fpm. In all cases the results were of the same order. Measurements were likewise made of the frictional resistance of the lining material. It was found that this material increased resistance by about 19 per cent more than the unlined duct, an increase which does not appear sufficient to account for the results.

It was concluded that this non-linear reduction was characteristic of the system. The next step was to test other systems for comparison purposes. Two such systems were investigated. These were exhaust ducts leading from other laboratory spaces, operating at a rather low air velocity. One system consisted of an 11 in. diameter round duct with a run of about 20 ft to the fan, containing three right angle bends. The other system consisted of a 6¾ in. diameter round duct, having a run of about 40 ft, with three right angle bends. Results on both systems were similar. Those obtained for the first

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system are presented here as typical in Table 7. (A different lining material was used, so the results are of a somewhat different order.)

Up to 512 cycles increasing length of lining produces an approximately proportional increase in attenuation, but above this frequency the same nonlinearity is observed as in the original experimental system. In all systems tested, this effect has appeared.

Sound absorbing treatment should evidently be placed at the outlet, since only in this way can it act upon sound generated as the air passes through the duct. For high frequency correction any reasonably efficient material is adequate and a length corresponding to ten diameters is probably ample. For low frequency correction and reductions in total loudness level a prediction can be based on the formula (6), but a factor of safety will be necessary until the effect of noise generated along the duct itself upon these lower frequencies is evaluated.

TABLE 7—REDUCTIONS IN NOISE LEVEL PRODUCED BY VARIOUS LENGTHS OF DUCT LINING

FREQUENCY	ABSORPTION COEFFICIENT OF MATERIAL	62 IN. LINING	124 IN. LINING	124 IN LINING AND BEND
0–64	...	0.6 db	1.6 db	2.6 db
64–128	0.10	0.6	2.6	3.2
128–256	0.13	1.8	4.3	4.7
256–512	0.30	4.2	8.6	8.2
512–1024	0.76	17.3	26.1	29.9
1024–2048	0.84	24.8	28.8	28.8
2048–4096	0.80	14.7	19.7	20.3

Since most sound absorbent materials of a porous nature are relatively inefficient at the low frequencies, other methods of sound elimination have received attention. This work has not progressed to a point where it can be reported, but it has already been found possible to produce many times the attenuation for a given length at low frequencies by specially designed wave traps and filters as compared with what can be obtained with simple lining material as ordinarily used.

FUTURE WORK

As stated earlier, this work can only be regarded as a progress report. Among the problems which still require solution are: (1) a final quantitative answer to the question of noise generated within the duct, (2) the effect of bends and splitters, (3) evaluation of the basis for the experimental constant in the formula, and (4) effect of grilles and standard outlet openings on the noise level. It is hoped that the discussion may inspire enough interest so that other investigators may see fit to carry on the work.

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DISCUSSION

PROF. A. I. BROWN: I believe the reason we have not had more discussion on these papers dealing with noise, is that we all know so little about it. It is such a difficult problem at times to undertake the investigation of noise. Frequently, in our engineering laboratories where we would like to undertake investigations, other noises are present to the extent that it makes it a very difficult problem.

Something that does not contribute to technical discussion of this but rather amused me, in connection with this paper, was the mention of permissible noises in railway coaches and Pullman cars. In the investigation of noises in Pullman cars, I have been told recently by some engineers that they were making a very careful study of noises, thought they were getting somewhere, measuring both in the daytime and at night, until some passenger let out a good husky snore and almost wrecked the meter.

O. H. WOLFF:[2] In recommending a slower speed motor, does that refer to a direct drive or would that also apply to a belt driven unit?

J. R. PATTERSON:[3] It would apply in either case.

D. J. STEWART: I have a question to ask. First, I would like to say a word about the subject which Professor Brown brought up, the difficulty of establishing any standard as to the permissible noise level. Strictly speaking, that is the province of the *American Society of Acoustical Engineers,* but this Society is so badly in need of the information that, perhaps, it would be possible for this Society to forward that research in some manner.

The figures given on Mr. Parkinson's slide, show residences, for example, as I recall, 25 to 35, the minimum was 25 decibels. We all know that is wrong, that is not the minimum at any rate, because a good quiet home in the country, with double windows and insulation is as low as 10 decibels on the 10^{-16} scale. That is, of course, when everything is otherwise quiet. You don't find many home fan systems, when running, that will run down to 10 decibels but, nevertheless, the base noise level is, in many cases, as low as 15, and I have measured at least two instances where the noise level was 10 or below.

Do you know whether or not those curves would have been the same shape for loudness levels, at 1000 cycles, for example, of 20 or 25 or 35 decibels? In other words, would the corresponding curve that would cut through here, would it have this same shape?

MR. PATTERSON: That I cannot say. There is this point to consider in the matter of noise, we don't have any definite technical data by which we can measure noise. There is so much of the human element involved, I could not even tell you about those two contours, as to what degree of noise they represent or very much about them. These curves of Mr. Parkinson's have been obtained elsewhere and they are, apparently, a valuable addition to our data because they give some idea of the relative annoyance of the different frequency.

I noticed in Mr. Stewart's paper some interesting straight lines which I have not had time to study, but the relation between your chart and this is apparent, and a study comparing the two should bring out some interesting points.

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[2] Engr., Dept. Public Utilities, City of St. Louis.
[3] Sales Engr., Johns-Manville Sales Corp., St. Louis, Mo.

MR. STEWART: I don't believe those straight lines on our chart have any bearing on this, Mr. Patterson, because they are the relation at constant loudness levels of the velocity and volume. They are merely a means of showing in a convenient form the effect of a combination of air and noise and they don't take into account the different loudness levels with the different types of noise. Our experiments were all made with a meter that had a frequency characteristic which is described in the paper on page 91, which, admittedly, we don't know much about, whether that is the correct shape or not, and in fact the new decibel meters have differences at different levels. Mr. Drake, I think, has looked into the new loudness meters a little more thoroughly than I have. Perhaps he can say a word about it.

G. F. DRAKE: The new meters, as they are being supplied now, conform with the Tentative Standards of the *A.S.A.* They provide three frequency weighting networks, one that gives a flat response for high loudness levels, one which is approximately correct for the loudness level contour at 70 decibels, and another for 30 decibels. These "loudness level contours" which I mentioned are supposed to represent the average response of the human ear to intensity levels of various magnitudes.

Now, I noticed in some of these charts that "loudness level" was referred to in some instances and "intensity level" in other instances. I was going to ask Mr. Patterson a question about that. I wonder if the measurements as taken were made with a weighted meter, that approximated the response of the human ear, or are they actually intensity levels. Can you tell me that?

MR. PATTERSON: No, I cannot tell you what the nature of the instrument is that was used in getting the data.

MR. DRAKE: Some explanation for the fact that most of the energy in the sound occurs in the lower frequency, is made by the fact that the response of the ear is less at those frequencies and that is greatest around 2000 or 5000 cycles per second.

MR. PATTERSON: It is a fact, in higher frequencies there is more annoyance, and we must consider reduction in those annoying frequencies of fans. In the matter of loudness level or intensity level, the intensity level is measured rather definitely. It contains none of the human element because it does not rely on the response of the human ear, whereas loudness level has practically all of the human element because we measure loudness levels more or less by response of the human ear.

In other words, there are two different scales there that we have to consider. That is not a very clear explanation of the two scales, but you must bear in mind the fact that the loudness level is measured, you might say, by the sense of hearing of various sounds, whereas intensity measuring has nothing to do with the human hearing, it is mathematical measuring of the energy of the sound.

MR. DRAKE: It was for that reason I was particularly interested in finding out if these decibel ratings, the figures which Mr. Parkinson's paper included, were on the basis of a weighted measurement; that is, loudness at that lower intensity level, which makes all of the difference in the world.

MR. PATTERSON: I cannot answer that, but I will be glad to find out for you.

MR. DRAKE: I cannot pose as an authority on sound. It so happens that in investigating this grille problem and the noise of air outlets, I found myself faced with the necessity of learning something about it. I will try to pass on a little of what I learned, but you can certainly go to more complete and reliable sources.

Noise, as you know, is detected by the human ear as a variation in pressure; pressure pulsations in the air. It is not an actual movement of the air from the

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source of sound to the ear, but a pressure wave that goes out from the source of sound and is picked up as a physical sensation by the ear.

Now, the intensity of the sound, as Mr. Patterson said, is a physical measure of the energy of the sound in this pulsating air wave and is not in any way dependent upon the way in which we hear the sound. Because of the big range of sound intensities to which the ear is sensitive, it has been found convenient to use a decibel scale in speaking of those values.

Now, the decibel scale sounds rather formidable and unless you are familiar with what it means, you are likely to get in some difficulties in making computations. It is on this basis: When we indicate intensity *levels* and loudness *levels*, as I will tell you in a few moments, we choose some threshold that will be a starting point or base value of intensity. Then we might say that we have one intensity of 100 threshold units—it does not make any difference what the units are,—and another intensity of 1000 units. Then, on the BEL scale, those intensity levels would be in the ratio of 2 and 3. In other words, the intensity level in bels is the logarithm of the intensity in terms of threshold units. The bel is the fundamental unit and the decibel is one-tenth of a bel. Therefore, the intensity level in decibels is ten times the logarithm of the intensity, so that the intensity levels would be 20 and 30 decibels.

You can see the tremendous range of intensity units which we would have to use to speak of intensity levels of 10 decibels and 100 decibels. The intensity would be in one case 10 and in the other 1,000,000,000 threshold units.

Now, so far as the response of the human ear to sound is concerned, we have to make some corrections for frequency, and because the human ear responds differently at different loudness levels, we have to make further corrections. The curve to which Mr. Stewart referred on page 89 is an approximate representation of the response of the human ear to an intensity which gives a loudness level of about 50 decibels at 1000 cycles per second. It is not accurate but it is an approximation which can be secured by a proper weighting network.

For the louder noises around 100 decibels the curve is substantially flat and that is the reason why the new sound level meter uses as one of its weighting networks, the absence of a weighting network; in other words, it measures the intensity level directly, because at 100 decibels the intensity and loudness level curves are almost identical when plotted against frequency.

As the noise goes down, the ear becomes less sensitive to the low frequencies than the high frequencies. Of course, it chops off the extremely high frequencies anyway, regardless of intensity, and down around the levels with which we ordinarily deal, 50 decibels, perhaps, this curve will give you a rough idea of the ear's response. That is, say, at 100 cycles the apparent loudness is down about 18 decibels.

The basis of all this, the threshold, should be something the most sensitive ear cannot hear. There is admittedly great variation in the sensitiveness of the ears of different people, and of the same person under different conditions, and consequently, any standards which are set up have to be based on a great number of observers.

The threshold which is now suggested as a tentative standard is 10^{-16} watts per square centimeter intensity. A noise of this intensity would be inaudible to all except the *extremely* sensitive ear. All of our noise measurements are on the basis of this threshold unit.

I would recommend a study of these Tentative Standards because in them is included a set of constant loudness contours, the result of extensive work by Mr.

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Fletcher of the Bell Telephone Laboratories. Also in the standards are recommendations for sound level meters which are based on certain of those constant loudness contours. 30, 70 and 100 decibel contours are duplicated as near as possible electrically in the meter so that the indication you get on the meter takes into account the variation in frequency of the source.

The way in which sound behaves in a room is something else. I have just been telling you the way you listen to it and the way we attempt to find out by means which we can depend upon to repeat just about how the average listener would respond to various noises. The intensity to which a noise will build up in a room depends upon the power which is going into the room from the source and the absorbing characteristics of the room. It is true that in some rooms—you probably have been in such—the walls are so reflecting, the room is so highly reverberant, that a noise will persist for some time; in large, highly reverberant rooms, this time may be a matter of seconds after the source is stopped. In such a room we have to be more concerned with the noise of grilles than we would in a room like a radio sound studio where the walls are highly absorbant; that is, if we were interested in the same sound level. However, they are much more exacting as to sound levels in the last named example, so the difficulty becomes greater although the absorbing characteristics of the room do help us a little.

A sabine is the coefficient of absorption of a material—this symbol alpha which appeared in Mr. Parkinson's paper—is simply that proportion of the sound incident on that surface which is not reflected; the difference between that and unity is the coefficient of reflection. Of course, this varies with different frequencies so the best we can do for an unpitched noise or general racket is to take some sort of an average coefficient of absorption. Such coefficients are given in THE GUIDE and in other sources, and if you know the area of that material which is in the room, and you multiply that area by the proper coefficient of absorption, you have the number of sabines which that material in that area contributes to the room.

If, for instance, the coefficient of absorption were 0.1 and the entire room of perhaps 10,000 square feet surface were covered with that material (which you are not likely to find in practice) the number of sabines in the room would be 1000. This is simply a measure of the total absorption in the room, and it is of importance in determining the reverberation time of the room for speakers, and how much noise we can get away with with grilles. Are there any other questions?

R. K. WERNER: As practicing engineer, I find at times difficulty in making fair and unbiased decisions or furnishing proof of such equipment as unit heaters, or unit ventilators, as to noise.

I believe it highly desirable that manufacturers provide in their data standard decibel ratings of their equipment. Some manufacturers have undertaken to do this but it is regrettable that their decibel ratings are not on an identical basis. I believe it highly desirable to incorporate in standardization of unit heaters, or unit ventilators, a standard uniform definition of decibel ratings.

MR. STEWART: There is a good deal of confusion on that very subject because there are three standards of threshold, three starting points for the decibel scale. However, the *Acoustical Society of America* has eliminated them all officially except the one that starts at an energy level such that zero decibels equal 10^{-16} watts per square centimeter. That is the only one that should be used hereafter, but in looking at tables and manufacturers' data you should be sure that the reference is to that threshold or that you know what the threshold is. Very frequently they are referred to another threshold that is about seven decibels above this value. In other words, you will find, according to a table of that sort, 30 decibels is the noise

of the particular heater, and that upon the accredited scale should be 37 decibels. There is still another one, isn't there, Mr. Drake?

MR. DRAKE: There is another one that is about 14 decibels higher.

MR. STEWART: In the course of the next few years those other bases for the decibel scale are going to be eliminated, I think.

E. K. CAMPBELL: I am not capable of discussing the matter from the standpoint of decibels. I am even more ignorant than Prof. Brown indicated, but there is one point that I wanted to bring out here, just possibly a misstatement, which was to the effect that the sound level increased with the increase of the speed of the motor. If that was a misstatement and the author meant the increase of the speed of the fan, I agree, but if he meant what he said, I don't agree at all. There is very little difference in noise given out by motors now at different speeds. I think that statement should be corrected if it was a misstatement.

MR. PATTERSON: I meant to imply fan speed as well as motor speed, not just, necessarily, motor speed alone.

M. D. KICZALES: A lot is heard about the noise and the elimination of noise in duct work. I have not heard any discussion about the prevention of noise by that piece of equipment known as the supply fan. There are lots of recommended speeds and offered velocities for different supply fans; but I want to know, and I believe a lot of the members would like to know, what is the best recommended offered velocity and tip speed of a fan which would give us the minimum amount of noise emanating from that fan.

W. A. ROWE: I am afraid we haven't time enough. A good many years ago Mr. Still told me, "Bill, you will find that there is a critical velocity where noise starts"—we didn't have decibels then and we didn't have sound measuring instruments —"somewhere around 3200 ft a minute, and if you will watch that, you will find it creeping into a lot of your work," and it has been interesting to note how it does crop out on different types of fan equipment.

J. S. PARKINSON: With respect to Mr. Stewart's comment on noise level in residences, I would like to point out that the figures given for noise levels in various locations are merely intended to be representative and cannot possibly cover all cases. It is altogether likely that a quiet home in the country might fall well below 25 db, as pointed out. On the other hand, it seems that it would be an unnecessary burden upon the ventilating contractor to require him to design a system which would be adequate for these extreme cases. It seems a more reasonable procedure to specify the average conditions which will be encountered, leaving it to the individual contractor to exercise his judgment as to where special precautions must be taken.

So far as information is available, it is believed that the annoyance curves would be approximately of the same shape at lower loudness levels. However, the data are not sufficient to warrant our drawing a curve at these levels.

In answer to Mr. Drake's question, the term loudness level where used in the charts refers to a suitably weighted measurement. The tests were made by measuring intensity level for each of a series of frequency bands. Where it was desired to obtain loudness level a correction was made at each frequency band in accordance with the ear sensitivity for that frequency and that intensity to arrive at the true loudness, and the total loudness level was obtained by summing the loudness levels at the individual frequency bands. The metering system used had a flat response

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characteristic in order to permit the initial measurement of intensity level to be made accurately.

I might add that I believe a slightly more accurate definition of threshold intensity would be to say that the threshold is that intensity level which is just audible to the average human ear. To an extremely sensitive ear such a level would be rather more plainly audible, whereas to an ear less sensitive than normal this level might not be audible at all.

No. 1053

INVESTIGATIONS ON THE EXCHANGES OF ENERGY BETWEEN THE BODY AND ITS ENVIRONMENT

By CHARLES SHEARD,* PH.D., MARVIN M. D. WILLIAMS,** PH.D., AND BAYARD T. HORTON,† M.D. (*NON-MEMBERS*), ROCHESTER, MINN.

THE constant internal temperature of the body is the result of the balance between the production of heat in the body and the loss of heat from the body. This constancy of internal temperature is maintained in the face of rather wide variations in the production of heat and in the environmental conditions. It has been shown by various investigators[1] that approximately 75 per cent of the total elimination of heat from the body is by radiation, conduction and convection from the skin and about 25 per cent by the vaporization of water from the skin and lungs when activities, short of physical labor, are carried on under comfortable environmental conditions. The necessity of the additional clothing in cold weather, on the one hand, in order to prevent an excess loss of heat from the body and, on the other hand, the inadequacy of the processes of radiation, convection and conduction to produce sufficient dissipation of heat in hot weather without increased vaporization of water from the skin are so well known that a passing mention of them is sufficient.

The temperature of the skin in any small area is the resultant of the heat brought to it largely by the blood and of the heat lost by the various dissipative processes which have been mentioned. Hence the amount of heat to be lost, in order to maintain constancy of the internal temperature of the body, and the temperature and humidity of the environment into which this energy has to be dissipated are the two chief factors which control the temperatures of various parts of the surface of the body. Recent investigations[2, 3] have indicated that, under increased environmental temperatures or with an increased

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† Consulting Physician, Division of Medicine, The Mayo Clinic.

[1] A Method for the Determination of Heat Production Over Long Periods of Time, L. H. Newburgh, F. H. Wiley and F. H. Lashmet, *Journal of Clinical Investigation*, 1931, *10*:703.

[2] The Rôle of the Extremities in the Dissipation of Heat, W. G. Maddock and F. A. Coller, *American Journal of Physiology*, 1933, *106*:589.

[3] The Function of Peripheral Vasoconstriction, F. A. Coller and W. G. Maddock, *Annals of Surgery*, 1934, *100*:983.

Presented at the 43rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, St. Louis, Mo., January, 1937, by Charles Sheard.

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production of heat, there is a change in the amount of blood, under the control of the mechanisms regulating temperature, which is brought to various portions of the surface of the body. Apparently, within the comfortable ranges of temperatures and humidities, this accuracy of regulation, in order that the production of heat may equal the loss of heat, is accomplished largely by the peripheral vascular systems of the extremities.

This presentation gives the results of investigations concerning the changes in temperature of the skin in various regions of the body produced under varied but controlled environmental conditions, direct evidence concerning the portions of the surface of the body which are chiefly concerned with the exchanges of energy between the body and its environment, as well as furnishing indirect evidence concerning the increase or decrease of supply of blood to various peripheral regions.

CONTROLLED ENVIRONMENTAL TEMPERATURES AND HUMIDITIES

In these investigations three psychrometric rooms were used, equipped with apparatus capable of maintaining the environmental temperatures within ± 1 F and the humidity within ± 3 per cent. In general, these rooms have been maintained at 65 to 68 F (17.5 to 20 C), 76 to 78 F (24.4 to 25.5 C) and 90 to 92 F (32.2 to 33.3C) respectively with a relative humidity of 40 per cent in order to investigate the effects produced by sudden changes in environmental temperature (accomplished by moving the bed and subject from one room to another) on the skin temperatures of various regions of the body in a basal metabolic condition as well as subsequent to the ingestion of food, exercise or change of posture of the body or of the extremities only. The maintenance of a room at a temperature of 77 F (25 C) and 40 per cent humidity the year around provides a means of investigating, under standard conditions, the skin temperatures of normal subjects and of those who have peripheral vascular deficiencies. Environments which are relatively warm (90 F) and cool (65 F) respectively enable these and other types of data to be obtained in cases of persons who complain of symptoms and reactions produced by heat and cold.

The subjects wore very light weight pajamas, so arranged that the arms and legs were exposed, and remained in a horizontal position on a comfortable bed during the period from 9 a.m. to 5 p.m. The head was raised slightly and a stand was arranged to carry reading material in order to minimize restlessness and to insure the arms remaining as much as possible in the horizontal position.

THERMOCOUPLES AND THE MEASUREMENT OF SKIN TEMPERATURES

The temperatures of the plantar surfaces of the right and left great toes (designated in Fig. 1 by the numerals 2 and 3), the volar sides of the distal phalanges of the right and left middle fingers (7 and 8), upper legs (4 and 5), ulnar wrists (9 and 10) and forehead (6) were measured by means of copper-constantan thermocouples. Thermocouple No. 1 was used for checking the temperature of the room in the vicinity of the subject or, in many instances, was a thermocouple so constructed and encased as to be suitable for the meas-

urement of the rectal temperature. Details relative to the construction of thermocouples and germane topics are to be found in a previous paper.[4] Each thermocouple was fastened in position by narrow strips of adhesive tape, the thermocouple leads (each 10 ft in length) being arranged and gathered into two suitable holders at the head and foot of the bed to assist in preventing looseness or poor contacts of the thermocouples. The distal ends of the copper

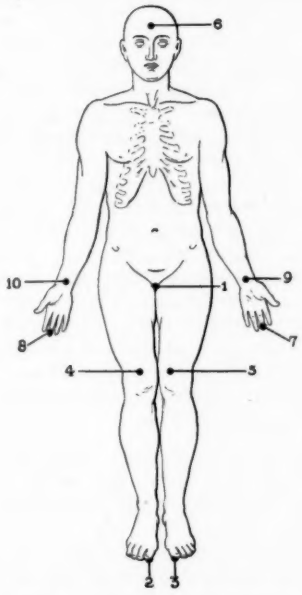

FIG. 1. DIAGRAMMATIC SKETCH SHOWING THE LOCATION OF THERMOCOUPLES

and constantan conductors are fastened to strips of copper and constantan which, in turn, are rigidly fastened into blocks of bakelite. These terminals are then plugged into a series of constantan-copper receptacles. By connecting the constantan receptacles in series with the constantan element of the common junction maintained at a specified temperature (42 C) and by using a bank of copper switches, it is possible to introduce any desired thermocouple into the galvanometric circuit. The deflections of the galvanometer, converted into equivalent degrees Centigrade, enable the investigator to obtain the skin temperatures very rapidly and with an accuracy of about 0.2 C. Similar sets of receptacles (20 in a room) are installed in each room and the readings of

[4] The Electromotive Thermometer: An Instrument and a Method for Measuring Intramural Intravenous, Superficial and Cavity Temperatures. Charles Sheard, *American Journal of Clinical Pathology*, 1931, *1*:209.

skin temperature are made by the operator at the central switchboard. Such an ensemble makes it possible to transfer bed, patient and thermocouples from one room to another, hence deleting the errors and delays occasioned by removing and replacing thermocouples. Data on several subjects can be obtained during the working period.

SKIN TEMPERATURES AND THE BASAL METABOLIC STATE

When the expression *basal metabolic state* is used, certain conditions are requisite, namely, fasting, normal body temperature, muscular relaxation and comfortable environment. The fasting, resting nude subject in an environmental temperature of about 75 to 77 F (24 to 25 C) with a relative humidity of 20 to 40 per cent produces heat at the basal rate. Since the internal temperature of the body remains constant it follows that the removal of heat from the body must be at the rate at which it is produced. It is necessary, therefore, to obtain data concerning the values of the skin temperatures in various areas of the body in the basal state in order properly to evaluate the rôle which the various portions of the surface of the body play in the dissipation of heat so that the temperature of the body, as evidenced by rectal readings, may be kept constant. Previous investigations[5], [6], [7] have shown that, in an environment of approximately 25 C and relative humidity of 25 per cent, the skin temperatures of the forehead, trunk and extreme upper portions of the arms and legs are commensurate and generally within the range of 32 to 35 C, whereas there is a temperature gradient in the extremities, the lowest temperatures being found in the fingers and toes.

A set of representative data obtained on a subject in the basal state is shown in Fig. 2. The room temperature was maintained at 77 F (25 C) with a relative humidity 40 per cent. Readings of the temperatures of the plantar surface of the left great toe only (curve 4) are given. Readings obtained under the same environmental conditions on the right great toe and other toes of this and other normal subjects have shown that the temperature curves closely parallel each other with temperature differences ranging between 1 and 2 C. The time-temperature curve of the ulnar wrist follows quite closely the curve for the finger, and is about 1 C lower at the corresponding time of measurement. The temperatures of the toes are maintained at a quite constant level, whereas the finer adjustments, or regulations of heat loss from the body, are controlled largely by the fingers and lower portions of the arms, as is evidenced by the rhythmically fluctuating curve of the temperatures of the fingers.

Curve 5 (Fig. 2) is the plot of the temperature readings of the toe of another subject occupying the room at the same time, and is introduced to show the differences in the temperatures of the extremities which may arise in individuals having nearly equal basal metabolic rates. The temperature of the forehead of subject No. 2 was maintained at 34.5, ± 0.25 C. The time-tem-

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[5] Loc. Cit. Notes 2 and 3.

[6] Human Calorimetry. III. Temperature Distribution, Blood Flow and Heat Storage in the Body in Basal Condition and After the Ingestion of Food, A. C. Burton and J. R. Murlin, *Journal of Nutrition*, 1935, 9:281.

[7] Changes in Surface and Rectal Temperatures in Man Produced by the Ingestion of Food Subsequent to Twenty-four Hour Fast, J. F. Herrick and Charles Sheard, *American Journal of Physiology*, 1935, 113:62.

perature data on the fingers and wrists show slight variations from 35.5 C. The temperatures of the forehead, fingers, wrists, and upper portions of the legs maintained maximal and approximately constant values; the dissipation of heat under the environmental condition specified was largely controlled in this person by the toes, feet and lower portions of the legs.

The initial portion of curve 5 shows a set of time-temperature relations for the toes which we have frequently encountered in these investigations when the subject has been up and around for an hour or more before undressing

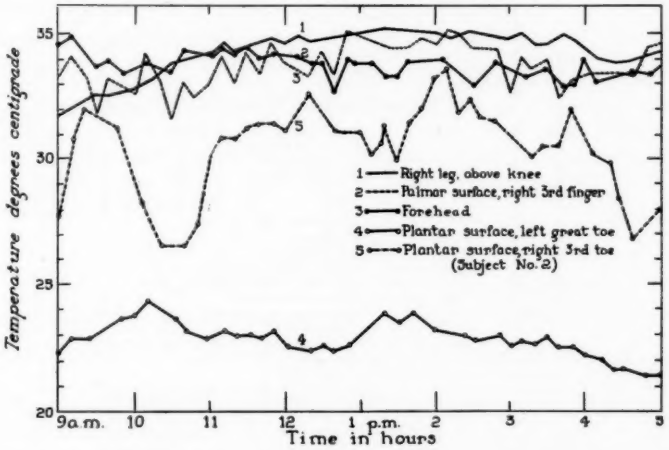

FIG. 2. CURVES (1, 2, 3 AND 4) OF THE SKIN TEMPERATURES OF A NORMAL MALE. CURVE 5 SHOWS THE TEMPERATURES OF THE TOES OF ANOTHER MALE OF THE SAME AGE, HEIGHT AND (APPROXIMATELY) WEIGHT. BOTH SUBJECTS IN THE BASAL METABOLIC STATE, ENVIRONMENTAL TEMPERATURE 77 F (25 C) AND RELATIVE HUMIDITY 40 PER CENT

and lying down on the bed. There is a rapid rise in temperature for the first half hour, followed by a corresponding decrease in the succeeding half hour, to be followed in turn by a rise to a value which is fairly high. These initial effects are evidently associated with change in posture and are the subject of a separate investigation.

It should be pointed out that the basal rates of these subjects were: No. 1, 43.7 cal per square meter per hour and No. 2, 41.1 cal per square meter per hour. They were both males, nearly of the same age, height and weight (20 lb difference). The dissipation of heat, as indicated by the measurements of the skin temperature, is considerably greater in the individual who has the lower basal metabolic rate. Attempts[8] have been made to correlate the average skin

[8] Loc. Cit. Notes 2 and 3.

temperatures of the great toes with the basal metabolic rates under constant environmental temperatures. The results obtained in these investigations cause doubt of the existence of such a linear relationship.

These tests definitely point to the necessity for obtaining the various skin temperatures over a considerable period of time while the subject is in the basal state, before proceeding to measurements of the effects produced by changes in environmental conditions, the ingestion of food, exercise, posture

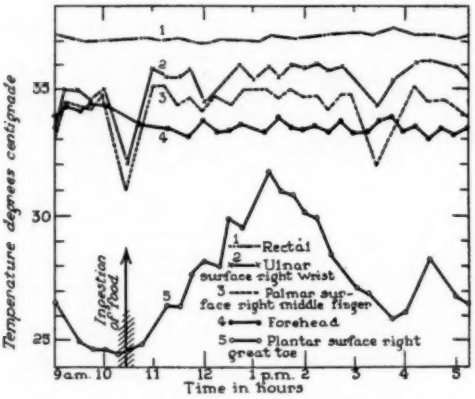

FIG. 3. CURVES SHOWING THE TIME-TEMPERATURE RELATIONSHIPS OF THE WRISTS, FINGERS AND TOES OF A NORMAL MALE SUBJECT IN THE BASAL CONDITION AND AFTER THE INGESTION OF FOOD. ENVIRONMENTAL TEMPERATURE 77 F (25 C) AND 40 PER CENT RELATIVE HUMIDITY

and the like or, again, in clinical investigations of individuals having vascular diseases.

THERMAL CHANGES INDUCED BY THE INGESTION OF FOOD

The necessity of basal metabolic conditions is emphasized when investigations are to be made on the so-called specific dynamic action of foods, or in the determination of the thermal changes produced in various areas of the skin of the body subsequent to the ingestion of food. After a fast of from 12 to 18 hours and after the temperatures in each area have remained constant to within approximately 1 C, the subject ate a meal which was balanced in essential types of food and sufficient to satisfy the hunger. In all instances, the temperature of the forehead remained constant within 1 to 1.5 C throughout the day. In general, shortly after the ingestion of food, the temperatures of the fingers, wrist and upper leg rise to their maximal values. The temperatures of the toes and of the lower extremities rise, in a room temperature of 25 C and 40 per cent humidity, in a manner represented in curve 5 of Fig. 3. The time

of the initial temperature change, the maximal value attained and the duration of the cycle of the change of the temperatures of the extremities vary with different individuals and in the same individual on different days. In general, it may be stated that the temperatures of the fingers and lower portions of the arms rise to their maximal values before the regulation of the elimination of heat is taken up by the lower extremities. There may be more than one cycle in the rise and fall of temperatures of the toes after the ingestion of food. In general, the blood pressures increased 5 to 15 per cent and the pulse rates were augmented 10 to 20 per cent shortly after the subject had eaten. These changes varied in their periods of duration, but, in general, decreased to normal values within an hour or two.

These results confirm the general trend of the conclusions recently reached in other investigations.[9] There are minor changes in rectal temperature following the ingestion of food. The changes in surface temperatures of the body, particularly in the extremities, subsequent to the ingestion of food are indicative of increase in blood flow or increased rate of circulation. These investigations concerning the superficial thermal changes in man find their counterpart and physiological explanation in previous work[10] concerning the effects of digestion on the blood flow in dogs.

EFFECTS ON SKIN TEMPERATURE

Changes of Environmental Temperature

After obtaining the findings on the changes in superficial temperatures produced in a group of normal individuals in the basal state and after the ingestion of food, investigations were made concerning the effects of changes of environmental temperature, the humidity remaining at 40 per cent, on the sites and manner of control of the loss of heat or its prevention as the case might be. Fairly characteristic results are diagrammed in Figs. 4 and 5.

The records of the measurements of skin temperature given in Fig. 4 clearly show the rôle played by the extremities, particularly marked in the fingers and toes, in the regulation of the loss of heat from the body, in the basal state, in order that the internal temperature of the body may be maintained quite constant at 98.6 F (37 C). The subject entered the room at 64 F (18 C) from an environmental temperature of approximately 78 F. At the end of an hour's time the temperature of the toes reached the temperature of the room. The marked regulation of loss of heat was accomplished by the fingers and lower portions of the arms, the temperatures of the fingers dropping from 34 to 22 C. The course of the changes of temperature of the wrist is similar to that of the fingers but not as great in range. On rapidly changing the room temperature to 68 F (20 C) the temperatures of the toes rose 1.5 C but were slightly below the room temperature at the end of 1.5 hours. There was a sudden rise in the temperatures of the fingers, followed by a lowering of the temperatures to a steady state approximating 25 C. The temperature of the wrist was fairly

[9] Loc. Cit. Notes 6 and 7.

[10] The Effect of Digestion on the Blood Flow in Certain Blood Vessels of the Dog, J. F. Herrick, H. E. Essex, F. C. Mann, and E. J. Baldes, *American Journal of Physiology*, 1934, *108*:621.

steady. When the environmental temperature was raised to 72.5 F (22.5 C), the temperatures of the fingers rose rapidly and were maintained in the range of 32 to 34 C, the wrist being at a temperature of 34 to 35 C. The temperatures of the toes rose in about an hour to a constantly maintained value of about 21.5 C. These results show that the toes assume the room temperature throughout the range of environmental temperatures (64 to 72.5 F), and that the loss of heat from the body is largely regulated by the hands and lower

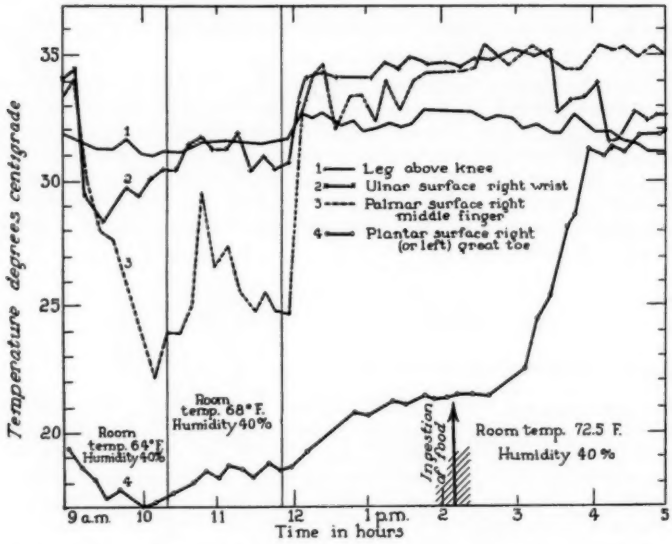

FIG. 4. CHANGES IN THE SUPERFICIAL TEMPERATURES OF THE LEGS, WRISTS, FINGERS AND TOES OF A NORMAL MALE IN THE BASAL STATE PRODUCED BY CHANGES IN ENVIRONMENTAL TEMPERATURE, THE HUMIDITY BEING MAINTAINED AT 40 PER CENT, AND AFTER THE INGESTION OF FOOD

portions of the arms. On repeated occasions this subject, in the basal state, showed temperatures of the great toes ranging from 31 to 33 C in an environment of 77 F (25 C), the humidity remaining 40 per cent. It seems possible, therefore, to establish quite definitely the conditions of environmental temperature at which the lower extremities begin to function actively as the regulators of the dissipation of heat from the body.

The ingestion of a standard meal showed the marked rise in temperatures of the toes from about room temperature (22 C) to 32 C.

The curves of Fig. 5 give the time-temperature relationships for the forehead, wrist, finger, leg and toe of another subject who, after fasting 15 hours, was placed in the room at 77 F (25 C) and, after 2.5 hours, was removed into an environmental temperature of 68 F (20 C). About 2.5 hours later

(total fast of 20 hours) the subject ate a meal; the data on superficial temperatures are plotted in the right-hand portion of Fig. 5. After the ingestion of food, the temperatures of the toes continued to fall and finally reached room temperature. The increased flow of blood to the fingers and wrists caused the marked rises in temperatures indicated, from which it may be concluded that, under the environmental conditions specified, the maintenance of the internal

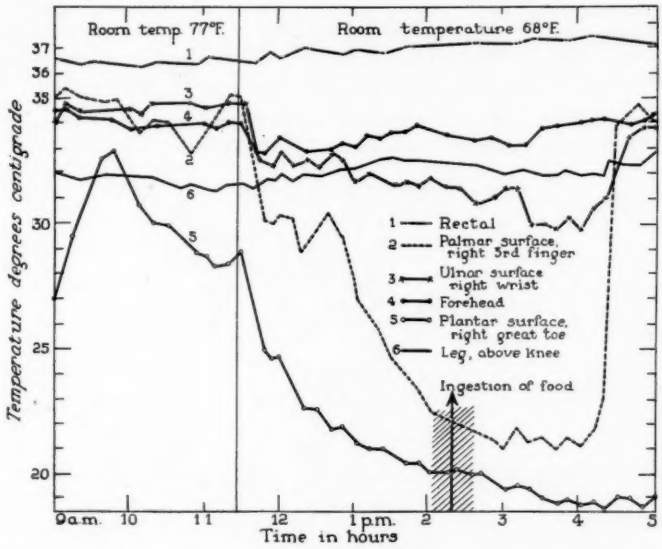

FIG. 5. CHANGES IN THE SKIN TEMPERATURES OF THE LEGS, WRISTS, FINGERS, TOES AND FOREHEAD OF A NORMAL MALE, IN THE BASAL STATE. PRODUCED BY CHANGES IN ENVIRONMENTAL TEMPERATURE, AND AFTER INGESTION OF FOOD. RELATIVE HUMIDITY, 40 PER CENT

temperature of the body was largely accomplished by the loss of heat from the hands and forearms. In general, these and similar data demonstrate the important rôle which the extremities play in the regulation of the dissipation of heat from the body under environmental temperatures ranging from 65 to 85 F.

Changes of Environmental Humidity

The effects of changes of humidity were investigated, the temperature of the room remaining constant, on the temperatures of various areas of the skin in a few individuals in an environmental temperature of 77 F, ± 1 F (24.8 to 25.5 C). Therefore, this portion of the discussion in particular should be considered in the nature of a preliminary account. A fairly typical set of time-temperature relationships concerning the thermal findings on the finger, ball of the great toe and the plantar surface of the great toe are given in the curves

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of Fig. 6. The subject, in the basal state, was placed in the room maintained at 70 to 75 per cent humidity. The temperature of the great toe remained at 25.3 ± 0.5 C for a period of 3 hours, the fingers and the wrists (not shown in Fig. 6) showing frequent cyclic changes in the range 31.5 to 34 C. The humidistat was then set for 30 per cent and the humidity dropped to the indicated value in about half an hour (at the fourth hour on the time scale). After remaining in the environmental humidity of 30 per cent for two hours, the humidistat was set at 75 per cent (at the sixth hour) and reached that value in half an hour. It is quite obvious from the data that the temperatures of the toes are several degrees (ranging from 2 to 4 C) lower in an environmental atmosphere of 30 per cent relative humidity. Also, the regulation of loss of

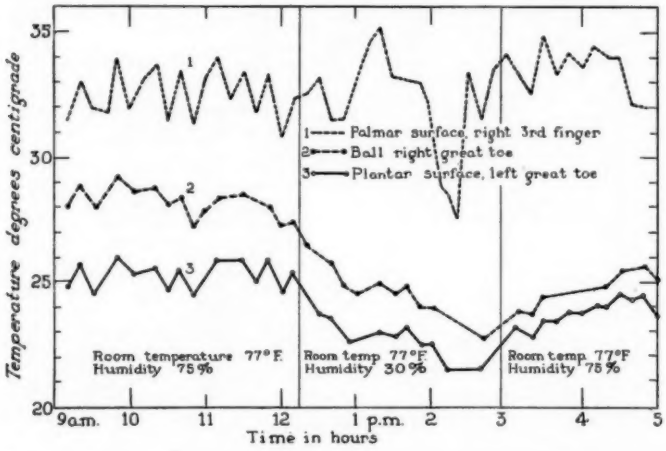

FIG. 6. CURVES SHOWING THE RELATIONSHIPS BETWEEN THE CHANGES IN SKIN TEMPERATURES OF THE EXTREMITIES AND CHANGES IN HUMIDITY, THE TEMPERATURE OF THE ROOM BEING MAINTAINED AT 77 F. NORMAL MALE IN THE BASAL CONDITION

heat by the fingers and wrists is more pronounced in low humidities. That the results are due essentially to changes in humidity is evidenced by the fact that the temperatures of the toes were elevated several degrees when the humidity was again increased to 75 per cent. Furthermore, data on the same subject, in a basal state, obtained on the preceding day under environmental conditions of 25 C and 40 to 45 per cent humidity, demonstrated that the temperatures of the great toes were 24.5 ± 0.7 C. The value 24.5 C fits in between the temperatures of the toes at 30 and 75 per cent humidity respectively.

It is probable that the increase of humidity causes increased temperatures of the feet and toes by reason of the fact that the normal loss (about 25 per cent) of heat by vaporization of water from the skin and lungs is markedly reduced.[11]

[11] Later investigations have indicated that relative humidity plays little part in the thermal exchanges, unless the body is in the zone of evaporative regulation.

SKIN TEMPERATURES IN VASCULAR DISEASES

In the preceding paragraphs, data obtained under specified and controlled environmental conditions on young healthy male subjects have been presented to show the important part played by the extremities in the control of dissipation of heat from the body. Increase or decrease of superficial temperatures is indicative of a corresponding change in the supply of blood to the peripheral tissues. In turn, the changes in the supply of blood are controlled largely by vasoconstriction of the blood vessels. Hence it follows that the marked regulation of the dissipation of heat by the extremities is accomplished by an equally marked and varying degree of peripheral vasoconstriction.

A discussion of the function of vasoconstriction and an application of the data and facts acquired by measurements of skin temperatures or other experi-

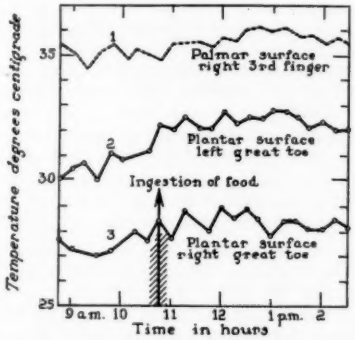

FIG. 7. CURVES SHOWING THE TIME-TEMPERATURE RELATIONSHIPS FOR THE FINGERS (EITHER HAND), RIGHT AND LEFT GREAT TOES IN A CASE OF PERIPHERAL VASCULAR DISEASE

mental procedures to conditions of peripheral vascular deficiencies are not within the scope of this paper. However, in order to contrast the type of data obtained by measurements of skin temperatures in vascular disease with the results obtained, under identical environmental conditions, from normal individuals, the time-temperature curves of Fig. 7 are included. These data should be compared with the findings shown in Fig. 3. The basal metabolic rates were practically the same in the two individuals. In both instances, the ranges and the courses of the time-temperature changes of the fingers, wrists and forehead were practically the same.

Under basal conditions, and in an environmental temperature of 77 F (25 C) and relative humidity of 40 per cent, Fig. 7 shows that the average temperature of the plantar surface of the right great toe was about 27.5 C, while the average reading on the left great toe was 30.5 C. These findings indicate differences in the heat-regulating functions of the two lower extremities due to varying

degrees of vasoconstriction or supply of blood. Following the ingestion of food there was a small increase (2 C) in the temperatures of the toes of both feet, thus definitely indicating the inability of the lower extremities to function as the regulators of the dissipation of heat from the body in the manner evidenced by normal persons under like conditions of environment and basal metabolism as indicated in Fig. 3.

The results of this series of investigations in general, as well as the findings of other investigators, show that, under certain environmental conditions, a portion of the so-called vasospasm is simply normal vasoconstriction, hence differentiation can be made only by studies of the peripheral vasoconstriction (as indicated by measurements of the skin temperatures of the extremities) in relation to the environmental conditions and by comparison with normal subjects under the same conditions.

SUMMARY

In this report there is presented evidence concerning:

1. The rôle of the extremities in the dissipation of heat from the body in order that the loss of heat may equal the production of heat, thereby maintaining the internal temperature of the body constant.

2. The influence of changes of environmental conditions on the temperatures of the skin in various areas in normal individuals in the basal condition.

3. The effects of the ingestion of food and the subsequent regulation of the dissipation of the increased production of heat by the upper and lower extremities of the body under varied but controlled environmental conditions.

4. The relative functions of the upper and lower extremities respectively in the control of the loss of heat from the body, under various controlled environments, in the basal state and following the ingestion of food.

5. The significance of the deductions, drawn from these investigations, concerning the function of vasoconstriction and the application of the experimental procedures which have been presented to the study of normal vasoconstriction and peripheral vascular diseases.

DISCUSSION

DR. E. V. HILL: Mr. President, I will at least start the discussion by saying how deeply I appreciate this paper by Dr. Sheard. It was about two years ago, when I first visited his Laboratory in Rochester, that I told him about the work I had done in determining comfort conditions for the body by the forehead skin temperature. He told me—he did it in a very courteous way—that I was "all wet," that the regulating mechanism of the body was with the hands and feet rather than the forehead, and it is something that we have to give some very serious thought to from now on.

The comfort of the human body does probably depend upon the skin temperature, but the skin temperatures of the hands and feet are the things that vary, and it is the temperatures of the hands and feet that we must measure in order to know what is going on in the body. So, when we say that the skin temperature is so-and-so, or a man is warm or he is cold, or is comfortable or uncomfortable in a certain air environment, we must know if his hands are cold, or if his feet are cold, or if his hands and feet are warm. So, asking if one is comfortable or not has very little meaning unless we consider in the picture the temperature of the hands and feet.

We must apply this excellent work that Dr. Sheard has been doing to our work

on human comfort in relation to the air environment and discontinue the unscientific practice of asking a subject whether or not he is comfortable.

DR. MUNSON: I would like to ask the doctor a question. Do these proportions apply on all races? Have tests been made on the Negro and the Eskimo, in the two limits of climate, with the same reaction?

DR. CHARLES SHEARD: I am answering this question with the statement that I do not know. Up to the present time we have not been able to investigate the question of differences that may exist in individuals of widely different racial characteristics. I know of no reason why the same general type of results as I have presented to you should not be obtained on any group of normal individuals who are placed in the same environmental conditions and under the same system of experimental procedure. As a passing comment concerning such people as Eskimos and Negroes, I have to say that the only Eskimos I have had were those of us who live in Minnesota, especially during some of our winter months. It is possible, however, that there are certain racial differences which exist when such people as Eskimos and Negroes live in their usual environments and that discomfort as well as readjustments of the heat eliminating mechanisms of such individuals occur when they are situated in what may be considered as unnatural or abnormal environmental conditions. We are investigating at the present time the question of whether a selected group of individuals, placed on the same regimen and tested under standard conditions of 77 F and 40 per cent relative humidity, will show any differences in the superficial temperatures of various body areas during the four seasons of the year. We may be able to report on this matter in another year. Also, it is quite evident to us that sex and age are important factors. Also basal metabolic rates and temperatures of extremities bear some relationship to each other (possibly a linear relationship), for the higher the metabolic rates the higher will be the temperatures of the extremities in general.

It is entirely possible that the relative roles played by various mechanisms (vasodilatation and evaporation of moisture) and extent of superficial areas involved in the process of elimination of heat are considerably different in people living in cold and hot climates, respectively, and that inability to readjust to changes in climatic conditions may be responsible for certain types of diseases.

Before any definite answers can be given to this question, it will be necessary for a considerable number of investigators to carry out a large number of tests and examine many individuals, for there is in biology that which we may speak of as a spread of normalcy, with high and low limits. For example, I just presented to you some data on two boys of the same age, same weight and height, whose basal metabolic rates are practically the same, but the temperatures of the toes of one subject are several degrees Centigrade lower than in the other subject.

Thus far our investigations indicate that the temperatures of the hands and feet of women, when placed under the same environmental conditions, are lower (especially in the feet) than in the case of men of about the same age. However, in general the basal metabolic rates are lower in women than in men of the same age; hence one would expect, from the facts I have presented in the main discussion, that vasoconstriction of the peripheral vessels would be more marked in the lower extremities. It may be true that the environmental temperatures which are satisfactory and pronounced to be comfortable need to be about 5 F higher in the case of women. As you will remember, no clothing (except pajamas with sleeves and legs cut off) was worn in these investigations; obviously the amount and character of clothing, exercise and so forth, are modifying factors in everyday life and make the solution of engineering problems all the more difficult. However, in general it would appear that somewhat warmer environmental conditions are more acceptable and also actually beneficial for women. Women are more prone to develop dis-

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turbances of the circulatory system of the extremities (such diseases as Raynaud's disease) dependent upon vasomotor dysfunction, whereas men are more likely to develop such conditions as arteriosclerosis. Possibly environmental factors, as well as differences in biologic function and normal sphere of activity in life, may play a part in the type of vascular disease which may arise.

As engineers handling material things and physical ensembles which can be accurately checked and controlled, I think it should be pointed out that, biologically, no such thing is possible. No two individuals are exactly alike. When we use the term "normal" we have no "check and re-check" and no such thing as a normal or a control in the fullest sense of the term. Hence, taking subjects who are clinically considered as normally functioning individuals, and following the same technic and procedures, we expect to obtain so-called frequency distribution curves when data are plotted to show the relationship between two attributes (such as height and weight), all other factors remaining the same insofar as is known. Without presuming to enter into any detail, I desire to point out that statistical methods, which have been found to be of great value in biology and medicine, will also have to be applied to the problems in which we are interested, after sufficient data have been accumulated.

J. B. STEELE: I have a 67-year-old friend in Winnipeg, Mr. Denike, principal of the John M. King School, and he never wears gloves while playing outside sports, no matter how low the temperature may be in Manitoba.

DR. SHEARD: When the pioneers and the forefathers of some of us here in the room came to the Northwest, they were accustomed to the severity of weather conditions. Doubtless it was a case of the survival of the fittest. Also, we know that the body becomes adjusted to changing conditions and it is probable that a thickening of the skin, especially those portions of the hands exposed to rough out-of-door usage, would develop the equivalent of insulation between the capillary beds and the outer environment. Also, they were heavy eaters and therefore had sufficient energy to maintain the requirements of the body; they were also accustomed to strenuous out-of-door work. We know from experience that many persons can go without gloves, use light-weight overcoats and build up a resistance or suitable adaptation to such weather conditions. Many of us are, figuratively speaking, hothouse plants and are the so-called white collar men of leisure; the successful farmer is an out-of-door farmer. We could become adjusted to these conditions if we adjusted slowly and as naturally occurs with our changing seasons.

There are wide variations as I have pointed out. I can cite the case of a woman from the South who complained of heat and who stated that she was very comfortable in a room at a temperature of 62 F and 40 per cent relative humidity, wearing a minimal amount of clothing. However, as I have already pointed out, there is a spread of normalcy. In this case, however, there were indications of abnormalcy. It is probable that the role played by the sympathetic nervous system is modified from the usual situation in such cases; that less heat is lost by evaporation of moisture (sweat) or that other portions of the body are at relatively lower temperatures than are commonly found.

COL. W. A. DANIELSON: Have these studies led you to any conclusions about blood pressure?

DR. SHEARD: I have said little or nothing about blood pressure. Blood pressures vary during the day, depend on posture, exercise, amount and type of meal and so forth. In general, in our investigations and under the experimental procedure adopted, we have found no significant changes. After eating a meal, and while lying prone, there is a 10 to 15 per cent rise in blood pressure (systolic pressure) which may persist for an hour or thereabouts and then slowly returns to its value

prior to the ingestion of food. There is also a tendency to a rise in pulse rate, varying from 5 to 15 per cent in the case of human subjects, following a meal. In dogs, the pulse rate rises during the course of digestion and may be double the rate found when the animal is in a basal condition.

DR. HILL: Some of the curves indicate, and I notice Dr. Sheard mentioned the fact, that the temperature of the feet, sometimes for hours, was lower than the temperature of the air. That is something for the engineer to think about and I would like to know if he has an explanation.

DR. SHEARD: I am not sure as to the answer. Some months ago I should have been willing to assert that such a situation could not exist and that errors in thermocouple readings must have occurred. However, all thermocouples have checked each other to within about 0.1 to a maximal difference of 0.2 C. It would seem as though a chunk of iron at a slightly lower temperature than that of the room would finally assume room temperature in the course of a few hours. However, in the case of several subjects we have found that the temperatures of the feet remained all day slightly below the environmental conditions (78 F and 40 per cent humidity). One explanation, which may account for the data obtained and phenomenon exhibited, consists in the condition of affairs in which the subject enters the room after having been exposed for some time to a very cold outdoor environment (zero degrees or thereabouts); vasoconstriction of the extremities would take place, peripheral blood would be minimal in amount and considerable heat would be lost from the extremities (*i.e.* feet in particular), with the result that the superficial layers of tissue would act as insulating layers. Under environmental conditions in which the room temperature and skin temperatures of the toes approximated each other, the thermal gradient would be very low and hence changes in temperature would be extremely slow. Furthermore, in the case of dogs, in which affairs are so arranged that the extremities are placed in a cold environment (0 C to 10 C) for a relatively short period of time, and when in the basal state, the temperatures of the toes or feet will drop from 35 C to about 20 C and remain at this lower temperature for two or three hours in an environment of about 25 C (77 F).

F. C. HOUGHTEN: I would like to ask if these low temperatures of the feet were lower than the wet-bulb temperature of the air.

DR. SHEARD: No: if the temperature is 75 F, the wet-bulb would be 58 F. They are not lower.

MR. HOUGHTEN: The evaporation would account for that?

DR. SHEARD: In certain types of disease, such as Raynaud's disease, the hands and feet are cold and frequently very moist. In an expressive fashion, they are said to drip water. There are, therefore, definite films of moisture and what has been suggested by you is undoubtedly occurring in such cases; that is, there is loss of heat by evaporation of abnormally large amounts of superficial moisture.

DR. HILL: This is a very interesting proposition because it means that the skin is evaporating moisture even though it is apparently dry.

DR. SHEARD: Correctly stated, I believe. In this room we are probably losing about 20 per cent of the heat from the body by evaporation. If the relative humidity is increased, the dry-bulb thermometer remaining constant, the temperatures of the extremities will be somewhat higher because loss of heat by evaporation is prevented by the increased humidity.

W. ROSS KNUDSEN: If I am correctly informed, there are nerve ends in one's arms and legs that respond differently, one to heat and one to cold. Could it be that over development of these nerve ends produces the different characteristics in individuals, as to their feelings of heat and cold? Have tests been made on individuals

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in extreme environments? For illustration, have the Eskimos of the frigid zone been tested, as well as the Negroes of the torrid zone, to see if their responses to these various characteristics are identical?

DR. SHEARD: You do get the response to nerve stimulation. That is presumably what happens in the so-called spastic conditions in which over-activity of the sympathetic nervous system produces marked vasoconstriction. The various nerve endings scattered over the periphery of the body are there for the purpose of transmitting nerve impulses initiated by nerve stimulation. These responses to stimuli from the external world are transmitted to a regulating center (or centers) and operate in the interest of the maintenance of normal function. The body is maintained at a nearly normal temperature; where the heat regulating center is located is not definitely known.

W. R. BEACH: Were the feet exposed or covered?

DR. SHEARD: Perhaps I should have been more explicit and given more details in my presentation. The subject removes his clothes and puts on a pair of shorts or sleeveless and legless pajamas. He lies prone and has no covering over him of any type. There are no drafts in the rooms, since this is an important point, for circulation of air in the form of drafts is an excellent way to cool the body and, under certain conditions, also an excellent method of contracting colds in the head. We have carried out tests with clothing and, while we have some interesting data, we are not ready to report the facts and their possible significance in the practical problems of air-conditioning.

D. J. STEWART: You called our attention to the fact that the temperature of the foot rises after a meal, now, why is it that many people, if not most people, feel cold especially in their feet immediately after a meal?

DR. SHEARD: Subjective evidence is extremely unreliable because it is often biased; there is no criterion of judgment, the subject has information which unduly influences him and so on ad infinitum. For instance, any information that I may give you as to whether I feel well or do not feel well is wholly dependent on my slant of mind, what is uppermost in my mind, what the trials or joys of life may be and so forth. We have no *objective* tests of comfort. If one says he is comfortable, what methods have we of ascertaining whether he is or is not comfortable or whether his criterion of comfort is a variable one. Many terms in the language are difficult of definition: such words as pain, happiness, well-being, discomfort. The factors which enter into the concept of comfort or discomfort are so varied that it is entirely possible that a zone of comfort is partially a myth, except in the sense that there is a lower and upper limit of temperature or humidity (or both) which produces physical reactions which are interpreted by the average person as uncomfortable.

A story that is appropriate at this point is taken from "Believe It or Not," by Ripley. A celebrated physician, Dr. Boerhaeve, left at his death an elaborately bound volume entitled: "The Onliest Secret of Medicine" which, by the way, is still being sought. This volume was sold at auction at the high figure of $10,000. When it was opened it was found to contain one page of material only. This page bore these words: "Keep your feet warm and your head cool, and you will keep the best doctor poor." The doctrine is simple and I suspect that the advice is quite wholesome, for I do believe that environmental conditions which permit the extremities (the feet and hands, upper and lower extremities) to be relatively warm are conducive to health and to the prevention of many a cold.

No. 1054

FEVER THERAPY INDUCED BY CONDITIONED AIR

By F. C. HOUGHTEN * (*MEMBER*), M. B. FERDERBER, M.D.,** AND CARL GUTBERLET *** (*NON-MEMBERS*), PITTSBURGH, PA.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

IT is recognized that the recent increase in longevity has resulted, in some measure, from sanitation applied to sewage disposal and the care of food, thus lessening the hazards of infection in the digestive tract. There is reason to believe that a similar extension may result in the future through similar sanitary applications to the atmospheric environment, or air conditioning.

Certainly when the human body is considered as a prime mover taking in food and air to be oxidized in the transformation of energy to be expended in physical activity, and in maintaining the required limits in body temperature under widely varying environmental conditions, it would appear that the control of this environment in order to lessen extremes in adaptation is desirable. No engineer attempting to control the temperature of any device or material in the process of heat treatment, where even a relatively large margin of temperature variation is permissible, would consider for a minute allowing the temperature of the environmental air to fluctuate at will over wide limits. Yet in disease little attention is paid to the environmental air to which body heat must be dissipated, even when disease most violently upsets the temperature control mechanism of the body. It is logical to believe that in many, if not all, human ailments the patient would react best under some optimum atmospheric surroundings. What air conditions should constitute these surroundings in any given case is not even suggested at this time, but the establishment of the particular conditions under which any given disease might react most favorably should not be difficult to determine, if once the air conditioning engineer and the medical man cooperate in studying the problem.

It has recently developed that certain diseases, including syphilis and gonorrhea, react favorably to an elevated body temperature. This was first observed

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when patients suffering one of these ailments contracted typhoid fever, and as a result were apparently cured of the social disease. Experimentation indicated that the rise in body temperature resulting from the fever produced the desired effect, and as a result artificially induced malaria fever has been resorted to as a cure. It has since been reasoned that if the rise in body temperature resulting from malaria fever is in itself the primary condition resulting in the cure, then other means for elevating the body temperature could more satisfactorily be used. Diathermy, or the raising of the temperature of the tissues of the body through the application of radio-magnetic waves within the body itself, has been used with some success. More recently, other means of elevating the body temperature of the patient have been used.

In 1929, a group of physicians and engineers, discussing the problem at the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, agreed that probably the most satisfactory means of uniformly elevating the body temperature without undue hazard and discomfort was through a proper control of the temperature and humidity of the environmental atmosphere. During the period from 1920 to 1929, studies [1] were conducted by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, in cooperation with the U. S. Public Health Service and with the U. S. Bureau of Mines at Pittsburgh, to determine the physiological reactions of men working in the high temperatures required in some industries. These studies established many facts regarding the rise in body temperature, increase in pulse rate, and other physiological reactions of such persons. As a result, there was made available a comprehensive record of physiological reactions of persons over a very wide range of relative humidity, temperature and air movement, including a range of temperature from the upper limits of the Comfort Zone to high temperature conditions in which life could not continue for more than one hour. From these records it is a simple matter to determine suitable combinations of temperature and humidity in either still or moving air to give any desired rate of rise in body temperature.

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[1] Body Temperatures and Their Measurements, by W. J. McConnell, M.D., and F. C. Houghten, A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 211.

Some Physiological Reactions of High Temperatures and Humidities, by W. J. McConnell, M.D., and F. C. Houghten, A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 122.

Further Study of Physiological Reactions, by W. J. McConnell, M.D., F. C. Houghten and F. M. Phillips, A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 353.

Air Motion, High Temperatures and Various Humidities Reactions on Human Beings, by W. J. McConnell, M.D., F. C. Houghten and C. P. Yaglou, A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 167.

Work Tests Conducted in Atmospheres of High Temperatures and Various Humidities in Still and Moving Air, by W. J. McConnell, M.D., and C. P. Yaglou, A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 101.

Body Temperature Regulation and Other Physiological Reactions in Their Relation to The Physical Characteristics of Man's Atmospheric Environment, by F. C. Houghten and W. W. Teague, A.S.H.V.E. JOURNAL, December, 1928, p. 873.

Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant, A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 245.

Heat and Moisture Losses from Men at Work and Their Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant, A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 541.

Thermal Exchanges Between the Human Body and Its Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant, *The American Journal of Physiology*, Vol. 88, No. 3, April, 1929.

Thermal Exchanges Between the Bodies of Men Working and the Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant, *American Journal of Hygiene*, Vol. XIII, No. 2, March, 1931, p. 415.

Fig. 1, taken from these records, shows the rate of rise in body temperature and the increase in pulse rate of three normal men in an atmosphere of 112 F, and with 100 per cent, 60 per cent, 30 per cent relative humidity. It will be noted that with 30 per cent relative humidity this high temperature results in a relatively small change in pulse rate and body temperature, while a saturated atmosphere with the same temperature gives rapid changes.

Following the first conference on the subject of fever therapy at the Research Laboratory, Dr. H. W. Wuerthele and the members of the Laboratory

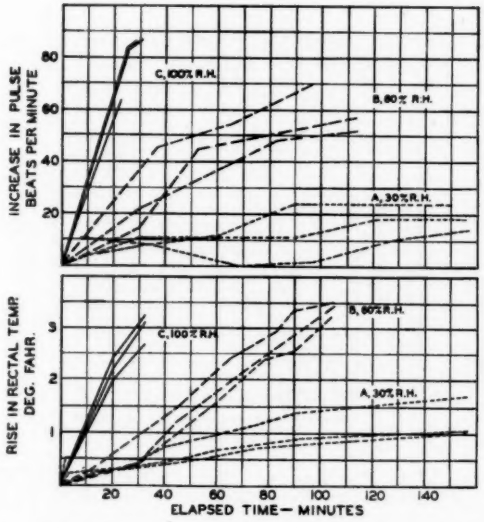

FIG. 1. RELATION BETWEEN RISE IN BODY TEMPERATURE AND INCREASE IN PULSE RATE FOR THREE NORMAL PERSONS AFTER ENTERING AN ATMOSPHERE OF 112 F AND 30, 60, AND 100 PER CENT RELATIVE HUMIDITIES

Staff agreed in March 1929 to the treatment of two cerebro-spinal syphilitic patients in the psychrometric rooms of the Research Laboratory. A series of fever treatments resulted in satisfactory improvements, demonstrating conclusively the feasibility of the application of air conditioning in the elevation of body temperature.

Early in 1935 another conference on the subject resulted in agreement that the best method of treating such cases was not in a large room, where several patients could be handled, but in a small box, where a single patient could be treated in a hospital under the control of the physician and nurse. Another question considered during this conference was whether the high temperature

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to which the disease organism is subjected in fever therapy is the primary and only factor in its destruction, or whether the rise in body temperature might not be accompanied by an increase, in the leucocyte count of the blood. Such an increase in the leucocyte count might be either the real cause of the cure, or a secondary factor. The earlier work at the Laboratory in cooperation with the U. S. Public Health Service had not revealed any consistent change in the leucocyte count of the blood.

A series of tests was planned and carried out in 1935 in the psychrometric rooms of the Research Laboratory, in cooperation with physicians of the

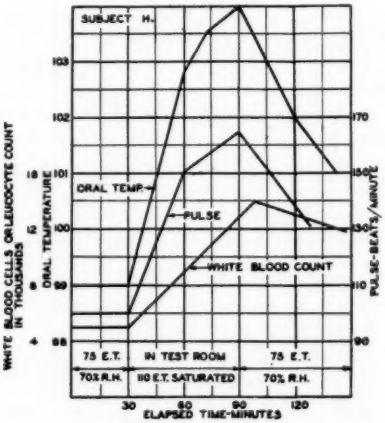

FIG. 2. RISE IN BODY TEMPERATURE, CHANGE IN PULSE RATE AND CHANGE IN LEUCOCYTE (WHITE BLOOD) COUNT, DURING EXPOSURE TO AN ATMOSPHERIC CONDITION OF 110 DRY-BULB, AND 100 PER CENT RELATIVE HUMIDITY

St. Francis Hospital. In these tests five persons offered themselves as subjects and were placed in saturated atmospheres of 105 and 110 F. In all but one of these cases, a phenomenal increase in the leucocyte count of the blood paralleled the rise in body temperature, indicating that the increase in the leucocyte count might be an important factor in the application of fever therapy. The relation between the hot condition, elapsed time and rise in body temperature, increase in pulse and increase in leucocyte count for two of the subjects are given in Figs. 2 and 3. The negative case, showing no material rise in leucocyte count, was a person of rather low vitality, who suffered unusual discomfort from the high temperature, and who collapsed when his body temperature reached 102 F. While this case proves nothing, it serves to indicate possibilities for further research.

After an analysis of the available data, in the files of the Research Laboratory, on the relation of rise of body temperature to the discomfort of persons subjected to hot atmospheres with different humidities, it was agreed that a most satisfactory box condition for giving fever therapy would be a saturated atmosphere. This would give a rapid rise in body temperature with a relatively low dry-bulb temperature of the air and a similar low temperature of all surfaces and objects in or around the box which might come in contact with the patient. Further, it appeared from the experience gained in the earlier studies that less discomfort resulted from a given rise in body temperature in a hot atmosphere with a high rather than a low relative humidity. Accord-

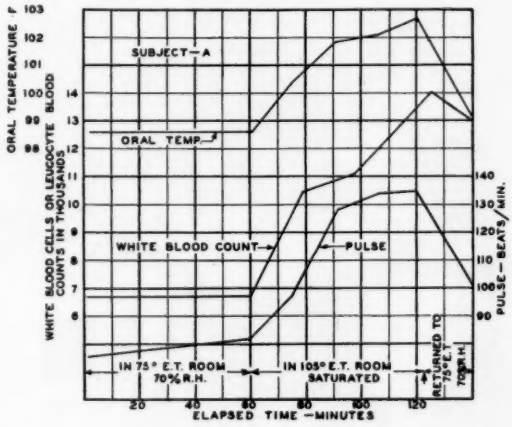

FIG. 3. RISE IN BODY TEMPERATURE, CHANGE IN PULSE RATE AND CHANGE IN LEUCOCYTE (WHITE BLOOD) COUNT, DURING EXPOSURE TO AN ATMOSPHERIC CONDITION OF 110 DRY-BULB, AND 100 PER CENT RELATIVE HUMIDITY

ingly, a careful study of the various possibilities in design which would give these conditions resulted in the box shown in Fig. 4. A drawing of the box, showing diagrammatically the arrangement of the air conditioning apparatus, is illustrated in Fig. 5.

The high temperature water spray serves to elevate the temperature and moisture content of the air, and also to circulate it through the system. The slight drop in air temperature from the point where it leaves the water spray to the box insures saturation in the box at all times when the spray is running. The water spray gives sufficient air velocity in the 6-in. duct to give a 115 cfm air supply, or 2½ air changes in the box every minute. The high temperature, humid air supplied at the top, and rapidly removed at the bottom, gives a very uniform air temperature throughout any horizontal cross-section of the box,

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and a temperature gradient from top to bottom of no more than 2 deg. The fact that neither a heater nor any other object or surface is ever found in the box at a higher temperature than the circulating air insures against burns from any imaginable accident or carelessness in operation. Since saturated air even at a considerably higher temperature than that used will cause no damage or disturbance other than an increase in the rate of rise in body temperature, there need be no fear of harmful effects from this source.

FIG. 4. AIR CONDITIONED FEVER THERAPY BOX IN OPERATION

Essentially, the air conditioning equipment is extremely simple and of the dew-point control type. Any source of hot water will serve the purpose equally well. In the experimental studies at St. Francis Hospital, thus far conducted, tap water from the hot water supply service was used, thus requiring no heater or mechanically moving parts in the entire equipment. The fact that this service could be supplied most easily by the hospital seemed to justify its use, although from an engineering point of view the practice would be recognized as wasteful of both water and heat. In later use of the box, a pump has been applied for recirculating the water through an electrical heater. Either a gas-fired heater or steam heat exchanger would serve equally well. It

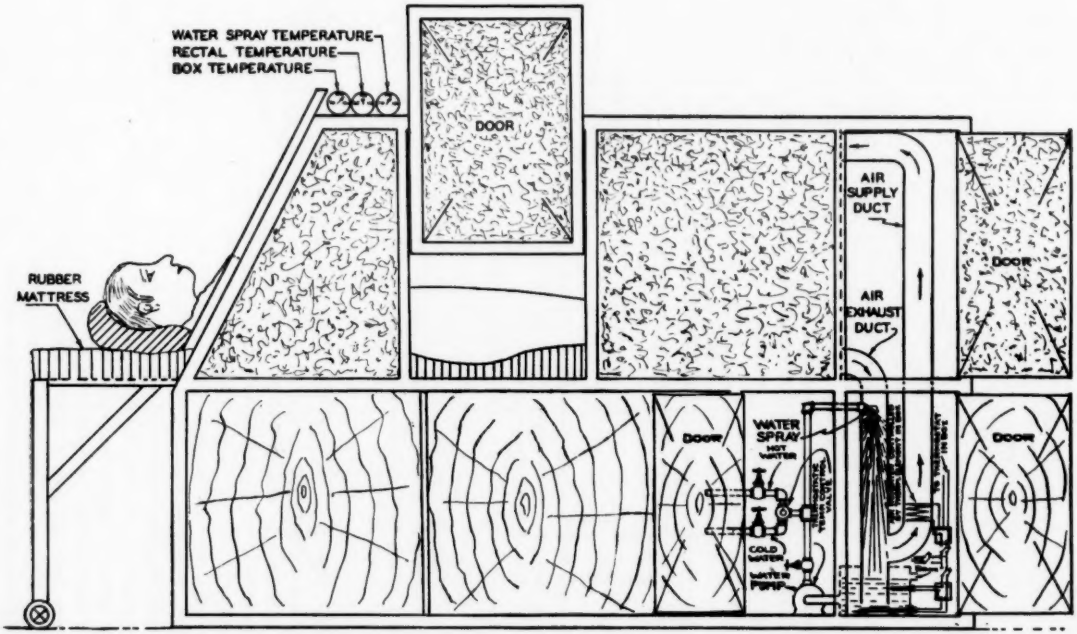

FIG. 5. AIR CONDITIONED FEVER THERAPY BOX SHOWING DIAGRAMMATIC SKETCH OF AIR CONDITIONING EQUIPMENT

has been found desirable to add a small amount of fresh water to keep down odors in the box.

Proper control of the air temperature in the box and of the body temperature of the patient requires ample facilities for control. Complete control may be had either by temperature indicating, or better, temperature recording instruments, giving the hot water temperature at the spray, the air temperature in the box, and the rectal or oral temperature of the patient. Suitable instruments of either the indicating or recording type are available for any of these purposes. In the experimental work thus far carried on, copper-constantan thermo-couples used with a precision potentiometer and mercury thermometers have been found more serviceable.

The patient is placed on the sponge rubber mattress and pushed into the box in a room at a slightly elevated temperature. The hot water spray is then

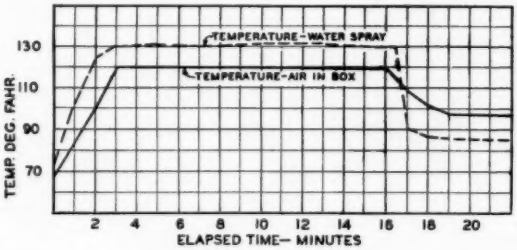

FIG. 6. RELATION BETWEEN SPRAY WATER AND BOX TEMPERATURE AFTER VALVE CONTROLLING 130 F WATER WAS TURNED ON *

* Avg spray water consumption 30 gal per hour. Avg air supply 115 cfm or approx. 2½ air changes per minute.

turned on. Fig. 6 shows the rise in the spray water temperature and the resulting box temperature on a trial run with no patient in the box and when the hot water supply was at 130 F. A very satisfactory operation of the box with a patient in it is as follows: After the patient is in the box, the hot water is turned on and the box temperature raised to 120 F. Under favorable circumstances, this can be done in 3 min. The box temperature may then be kept at 120 F for about one-half hour, during which time the patient's temperature may be expected to rise from normal to 104 F. The box temperature is then dropped to 110 F for an additional 20 min, which may be expected to bring the patient's temperature up to 106 F. If this is the degree of fever predetermined for the particular patient the box temperature is dropped to a temperature between 100 to 103 F, so as to maintain the patient's body temperature at 106 F. After the patient's temperature has reached the desired level, a lower air temperature in the box is required, due to the fact that the patient himself is generating in the neighborhood of 400 Btu of heat per hour, which, in order not to raise his body temperature further, must be dissipated to the atmosphere in the box.

It is not the aim of the authors to present a complete professional record of the treatment of patients in this engineering report. It is hoped that such a record will be published in the medical press. However, the following brief discussion of the subject may be of interest.

The clinical application of fever therapy induced by air conditioning to the treatment of disease has been both interesting and satisfactory in most cases. Up to this time, 70 patients have been treated. Gonorrhea, one of the most common of venereal diseases, has been treated in almost all of its phases. The organism, Neisseria Gonorrhoeae, has a thermal death point between 106 and 107 F in vitro, but workers previously have shown that if the patient's temperature is elevated between these points, the organism is destroyed similarly. In addition to this, elevating the temperature stimulates the body in such a way that the white blood corpuscles are increased in number. These act as scavengers, which clean up debris and inflammatory material left by the infection caused by the organism. This is especially true in females with gonorrhea, who complain very bitterly of severe pain in the lower abdomen. Because of the peculiar structure of the female pelvis and the location of the organs, there are many areas which become filled with this inflammatory material, binding down the organs in such a way that pain always occurs. It is very gratifying to see these patients get relief even after one treatment of about one to three hours, and with a full course of treatment their pain recedes and does not recur unless they become reinfected.

In the male, the relief of urethral symptoms such as pain, burning, etc., is rapid with Fever Therapy. The use of this type of treatment prevents complications which leave their stigmata in the male. However, in both sexes arthritis caused by gonorrhea can be treated with fever therapy *specifically*. If treated early, there is no joint disability, and the patient is restored to usefulness. In chronic cases where some joint destruction has occurred, fever therapy frequently aids in restoring some function to the involved joint. When the excruciating pain of these arthritics and the striking relief obtained with fever therapy are observed, it is difficult to be skeptical about its usefulness. As in any other acute disease, the sooner treatment is instituted, the more gratifying are the results. Experience indicates that 12 to 25 hours are required to clean up this infection with sessions of 3 to 5 hours' duration given at 3 to 7 day intervals.

In syphilis the situation is entirely different. The organism *Treponema pallidum* attacks certain organs of the body, namely, the circulatory system (heart, blood vessels) and the central nervous system. In the latter connection, the brain is so attacked that various degrees of insanity and other cerebral manifestations result. The term paresis is applied to this ailment and it has very frequently been called *Insanity of Syphilis*. It is a degenerative disease with mental deterioration resulting in peculiar behavior on the part of the individual. There are many other manifestations of this disease in the nervous system, which are not discussed in detail in this paper.

Fever therapy arrests this degenerative process in a great majority of cases and frequently restores its victim to usefulness. Since the thermal death point of the organism varies from 103 to 106 F, it has been the practice in this study to maintain the patient's temperature at about 105 to 106 F. Malarial inoculation to produce fever has been in use for years. It is felt as a

result of this study that artificial fever has the additional advantage of being safer to handle without imposing one disease on another. It is hoped further study will show the relative ease of artificial fever administration as compared to malarial inoculation in cases of syphilis. The other manifestations of this disease may be treated with fever together with the various chemicals, and good results have been reported. The combination of fever therapy and drug is very useful because the drug is disseminated more thoroughly through the tissues because of its easy passage through the blood vessels dilated by the fever. In paresis about 25 to 50 hours produce the best results with 5 hour sessions given every week.

In Chorea, or St. Vitus' Dance, fever therapy has done much to bring about recovery and shorten the course of the disease. Even after the first treatment, the purposeless, involuntary movements abate and frequently cease. Eight or nine sessions at 105 to 106 F for 3 to 4 hours are suggested in these cases. Since this particular disease has been treated more recently, there is not much data available on just how the method works. Some time in the future more may be written on this subject.

In this study fever therapy has been tried in multiple sclerosis, a degenerative disease of the central nervous system, without striking results, although other workers' reports are encouraging.

There has been some success with one case of Parkinson's Syndrome, a manifestation of encephalitis. This patient had a tremor of the right hand for 9 years and was unable to earn his living, but after 25 hours of fever between 105 and 106 F, this tremor has ceased, and he is now actively engaged in making a living. However, this single case proves nothing, but must be amplified by more cases which are now being studied.

Before accepting any case for fever, a case history is taken, the patient is carefully examined, especially for a weak heart, high blood pressure, or hardened arteries, all of which are contraindications to giving fever. Patients who are weakened and debilitated do not receive treatment. The blood counts, as well as bacteriological studies, are made before starting the procedure. A sedative is administered, and while lying in the box, the face and head are cooled both by an electric fan, and ice applied directly. This gives some degree of comfort to the patient and minimizes the danger of delirium. Since the chloride and fluid loss is great, 0.6 saline solution is given by mouth. Rectal temperatures, pulse and respiratory rates are taken every 10 min and blood pressure readings are made at greater intervals. The pressure is at first elevated and drops down as dilatation of the vessels takes place. No covers are used on the patient during treatment, thus permitting freedom of movement at all times. If the patient becomes restless, a small dose of opiate is administered. After the fever has been given, the patient is put to bed, and glucose and saline solution, which are given by vein, act as a stimulant and food, especially since the patient has eaten nothing prior to and during his treatment. Many patients come in the morning, have treatment, sleep a few hours following treatment, and leave the hospital in the evening without any difficulty whatsoever. Occasionally some individuals develop herpes (fever blisters) on the lips which at times become troublesome, but little can be done about it. The complication of fever delirium rarely occurs. Cyanosis occurs infrequently but the administration of O_2 (90 per cent) and CO_2 (10 per cent) clears this up imme-

diately and the treatment goes on. There have been no burns in this cabinet, and it is felt that this is quite a remarkable record when it is considered that these patients have no covering except a very short gown worn for modesty's sake. The following case records may be of interest:

A girl, 18 years old, was forced to leave high school in her senior year because of mental changes and some difficulty with her vision. She was referred to a well-known neuro-psychiatrist in Pittsburgh, and sent to the hospital. Laboratory data revealed the fact that she was suffering from congenital (hereditary) syphilis. Malarial inoculation was given to induce a fever, but nothing favorable resulted. Artificial fever in the Laboratory's box was then tried as the first case in this study.

After 12 hours of fever, varying from 105 to 106 F, her eye findings were greatly improved and her mutterings and mental aberrations ceased. Her responses were normal in every respect. She was discharged from the hospital, and has since returned to school and completed her studies.

An 18 year old colored male was carried into the hospital suffering from excruciating pains in the left ankle, knee, elbow, wrist and joints of the hand. Also affected were the right knee, elbow, wrist and hand. There was also a urethral discharge which upon laboratory diagnosis was revealed as gonorrheal. He was exposed to the infection 5 days before admission and the pain began 2 days before admission. Morphine was required to ease the pain and he was unable to use his hands and had to be fed. Fever was begun the next day, with the temperature between 106 and 107 F, for a 4 hour period. It must be remembered that this occurred during the extremely hot weather of July. After the first session, he was able to sleep without the aid of opiates and could handle his food. The treatments were repeated every 3 or 4 days and after 20 hours he was entirely clear of pain, upon either active or passive motion. At the same time the urethral discharge ceased and he was no longer kept in isolation. Three days later he was discharged from the hospital, and during the week of September 24, 1936 a complete fixation test (diagnostic test using patient's blood) was negative. Before treatment it was positive. In sharp contrast to this case was that of a young adult admitted to the hospital last winter with the same ailment only in two joints, the hip and knee. Without using fever, he remained in the hospital for 128 days, receiving such treatment as vaccines, diathermy and a cast on his body. The first case cited was in the hospital only 17 days. Since both were free cases, it is obvious that a great economic saving is possible with this form of treatment.

A 25 year old woman was exposed to gonorrhea and was brought to the hospital complaining of severe pain in the lower portion of the abdomen. Although a tentative diagnosis of appendicitis was made, the history, along with the laboratory findings, definitely indicated gonorrheal salpyngitis (fallopian tube inflammation). Fever was given between 106 and 107 F for a total of 18 hours, resulting in complete subsidence of pain and other symptoms. It is interesting to note that this patient could not rest at night because of the severe pain in the abdomen, and opiates were necessary to give the patient some measure of relief. In this instance, as in others, the acute pain subsided after the first treatment of 3½ hours, and the patient was able to sleep without the use of sedatives or opiates. This patient, however, developed herpes (fever sores) on her lower lip, which retarded treatment for 5 days. How-

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ever, treatment was resumed, and before discharge from the hospital, a vaginal examination revealed no pain or discomfort; nor were there any signs of abnormal changes in the pelvic organs. This latter complication frequently occurs if the condition is untreated, and an operation is the usual sequel in time. Hence, early treatment is desirable.

ACKNOWLEDGMENT

The authors wish to acknowledge the valuable aid rendered by: Dr. W. H. Wuerthele, whose interest in the possibilities of fever therapy, and whose foresight in recognizing the possibilities of applying air conditioning to the problem, originally encouraged the authors to undertake the study on the present basis; Sister M. Thomasine, Superintendent, and her staff of St. Francis Hospital, in making available the many facilities necessary for the study; Dr. A. W. Sherrill, Associate Professor of Medicine, University of Pittsburgh, and head of the Department of Physical Therapy at St. Francis, in whose department the work was carried on, for his systematic cooperation; M. J. McCarthy, Chief Engineer, who supplied the many mechanical facilities necessary for the successful operation of the box in the hospital; Miss Gertrude Demsko, R.N., who was constantly with the patients treated, and whose untiring efforts even after hours was a large factor in the success of the study.

DISCUSSION

DR. CHARLES SHEARD[2]: Mr. Houghten has made a very delightful and clear presentation of his material. For many years I have been interested in the field of fever therapy, as I am sure Mr. Houghten knows by reason of the various contributions which I have made to the literature of physiology and of clinical investigation concerning the production of fever by various physical and chemical methods and procedures. It is a pleasure to me to make some remarks concerning the biophysical principles and also to point out some physiologic and therapeutic effects of fever therapy.

It is common custom to trace the therapeutic value of heat to the Romans and Greeks. The establishment of bathing pools, however, cannot be regarded as therapeutic procedures; they smack of ordinary hygiene and cosmetics. True fever therapy as it is known today, originated with Wagner-Jauregg in 1918. He found that inoculation of malaria into a patient with paralytic dementia was frequently followed by improvement or complete remission of symptoms. Malaria therapy has been in considerable vogue during the past two decades; varying results have been reported due, undoubtedly, to the facility with which malaria therapy could be used under ordinary conditions. Various methods of producing fever have been applied to the treatment of a considerable number of diseases; clinical experience has indicated that induced fever is especially useful in the treatment of gonorrhea and syphilis.

Some years ago Dr. W. R. Whitney, then director of the research laboratories of the General Electric Co., made the observation that the mouth temperatures of men working near a short wave radio transmitting station were raised a couple of degrees (approximately) and that an induced or low degree artificial fever had been produced. These observations suggested the possibility of the use of short electric wave in fever therapy and gave a considerable stimulus to such investiga-

[2] Director, Division of Physics and Biophysical Research, The Mayo Foundation and The Mayo Clinic.

tions by reason of the fact that a physical method, which was relatively simple and could be controlled, was made available to the medical profession. In the use of the first types of short wave apparatus it was found that superficial burns were often produced. Subsequently engineers developed equipment which was based on electromagnetic instead of electrostatic coupling, the production of heat in the body being the result of an induced electromotive force and hence of eddy currents of the same frequency as the exciting current (of the order of 12,000,000 cycles per second) which flow in the conductive material. The inductotherm, as first developed and used, was announced in 1934 by the General Electric Co.

During the period of the past 10 years other investigators have developed other types of equipment for the production of fever. Carpenter and Warren of Rochester, N. Y., built cabinets carrying incandescent lamps. At about the same time C. F. Kettering and Dr. Walter Simpson of Dayton, O., developed a cabinet (known as the hypertherm) for hyperpyrexia in which circulating hot air, with satisfactory relative humidity, was employed. Medical literature is replete with information regarding physical ensembles and therapeutic uses of artificially induced fevers and should be consulted by engineers who are interested in the development of equipment suitable for the use of the medical profession. In particular, I call attention to the contributions of Bierman, Carpenter, Hinsie, Krusen, Neyman, Schliepake and Simpson.

The value of heat as a physical means of alleviating or curing diseases of certain types has been emphasized at various times throughout the history of medicine. However, the significance of fever and its relation to the course of an infectious disease or to the healing of trauma have been questioned. It has been believed that a rise in temperature was a sign of disease and that, to establish the comfort of the patient and to minister to his welfare, this increased body heat should be dissipated. During the last few years, however, evidence has been accumulated to indicate that a fever, other than hyperpyrexia of central origin, may be of value to the diseased body. It is now generally agreed that fever is essentially a defensive and protective mechanism. While all the factors involved in artificially produced fever are not known in all probability, and while satisfactory interpretation cannot be made at present, the chief manifestation lies in the physiologic effects produced by an increased amount of heat in the body for a given period of time. Through the work of various investigators it has been shown that the fever which accompanies infection exerts an adverse influence on the growth of bacteria, favors phagocytosis, diminishes the potency of toxins and stimulates the development of immune bodies. In the light of present conceptions regarding fever, we should consider fever as a natural, active factor produced within the body to aid in the destruction of an infectious agent and to repair tissue injury from any cause.

As biophysicists, considering the possible affects of increased temperature on the body, we divide the possibilities into three groups. The first group of possibilities has been mentioned, namely the probable destruction of foreign or parasitic life caused by the intolerance of bacteria and infectious agents to temperatures which the human body and its organs may withstand with safety. The second possible effect is the influence of heat on chemical reactions. In general, the speed of chemical reactions is increased with rise of temperature. Without doubt these chemical reactions are very extensive and complex. The third division lies in the effects of temperature on solubilities. Whitney, who has done much to stimulate research on clinical applications of artificially produced fever, stated that, in addition to the speed of life reactions, there is the entirely different factor of the partial solubility or insolubility of numberless substances which take part in the mobile equilibrium of life.

There are five general methods for producing fever: hot water and heated air baths, radiant heat (as from incandescent lamps), foreign proteins, infectious agents,

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and the use of high frequency energy (diathermy and radiothermy). Fundamentally, all of these methods depend on heat, which must be produced by and within the body or supplied from an outer environment of energy that is transformed into heat when applied to the body. All of these methods have some specific advantages and defects. From the physical standpoint, the rapidity with which the temperature of the body may be raised is dependent on the insulation against losses of heat and the rate of absorption of energy and its transformation into heat on the part of the body. In fever produced by external heat, the temperature gradient is from the outer environment into the body, the environmental temperature being that which can be withstood by the skin of the body. In radiothermy (short electric wave oscillator) the energy of the field induces electric (eddy) currents within the body which produce heat. By this latter method the heat is produced directly within the body, just as occurs in the course of a fever due to an infectious agent. Such a method permits internal heating in which the heat is generated in the organs of the body, heat being produced at a greater rate in the more conductive than in the less conductive tissues. Benson, working in Carlson's laboratory, carried on some investigations concerning the relative influence of external and body temperatures on the heart and concluded that the heart rate varied directly as the temperature applied to the skin of the body. From these and other experimental findings he argued for the use of methods for the production of artificial fever in which sufficient heat can be introduced into the body while the skin is kept comparatively cool. The experiments of Pratt and Sheard, using anesthetized animals, have shown that, by suitable procedures with radiothermy, muscle tissues are affected as to temperature more than subcutaneous tissue and that temperatures within joints are affected more than in muscle tissue.

In a general way, it may be concluded that: (1) the conception that fever is a destructive process only is being supplanted by proofs that fever is a defensive and protective mechanism; (2) the value of induced fever is being established in certain types of diseases; (3) whether the effects of radiothermy and diathermy (penetrating heat) and other means of producing artificial fever (applications of external heat) are comparable in all respects is a question, and (4) the similarity of clinical results that have been obtained by a variety of fever-inducing agents argues in favor of the premise that the results which have been obtained are attributable chiefly to the production of fever.

F. C. HOUGHTEN: There is no thought that the engineer is going to use the fever therapy box. It must be handled entirely by the medical man as a part of physical therapy treatment in a hospital. The conception of the use of this box and treatment of the disease was not that of the Laboratory. They simply supplied a means to get the rise in temperature. We acted purely as heat engineers.

No. 1055

COOLING REQUIREMENTS FOR SUMMER COMFORT AIR CONDITIONING

By F. C. Houghten,* Pittsburgh, Pa., F. E. Giesecke,** College Station, Tex., C. Tasker,† Toronto (*MEMBERS*), and Carl Gutberlet,‡ Pittsburgh, Pa. (*NON-MEMBER*)

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

IN the Spring of 1935 the Research Laboratory of the American Society of Heating and Ventilating Engineers was asked to make a study of the cooling requirements for summer comfort air conditioning. As a result of a brief study made in Pittsburgh during the summer of that year, a report [1] was presented to the 1936 Annual Meeting of the Society, indicating that the effective temperature scale was the best index of the relative feeling of warmth experienced by persons in summer air conditioned space. An effective temperature in the neighborhood of 73 deg was indicated as desirable in Pittsburgh under such conditions, rather than the 66 deg effective temperature generally accepted for winter heating and air conditioning. This study also gave information on the cold shock experienced upon entering a cooled space and the time required under different atmospheric conditions, for establishing comfort and for the disappearance of sensible perspiration.

The brief period available for the study, together with the small number of subjects and their age limitations, affected adversely the conclusiveness of the study, and the Technical Advisory Committee on Comfort Requirements for Summer Cooling, W. L. Fleisher, *Chairman,* A. E. Beals, F. R. Bichowsky, Thomas Chester, F. E. Giesecke, Elliott Harrington, Dr. E. V. Hill, R. E. Keyes, Cyril Tasker, C. P. Yaglou, agreed that the study should be continued.

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† Research Fellow, Ontario Research Foundation, Toronto, Ontario.
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[1] Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet. A.S.H.V.E. Transactions, Vol. 42, 1936.

Presented at the 43rd Annual Meeting of the American Society of Heating and Ventilating Engineers, St. Louis, Mo., January, 1937, by F. C. Houghten.

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The continued study was planned so as to give added information on the following points:

1. Desirable atmospheric conditions as regards temperature and humidity in still air for comfort in summer air conditioning, together with any variation in these requirements for different geographical regions having different weather conditions.
2. Allowable variations in the relative humidity of the air with optimum conditions of effective temperature, and any variation in the time required for the disappearance of sensible perspiration with variations in the moisture content of the air.
3. Variation in the required effective temperature for comfort depending upon the age, sex and general health conditions of the persons involved.
4. The effect of the amount of clothing worn, particularly the effect of coats worn by men, on the desired effective temperature.
5. The effect of air velocities, above those usually used in air conditioning, on the cooling requirements.
6. The characteristics and seriousness of the cold shock felt upon entering a cooled space in the summer, together with the duration of the shock and the subsequent physiological reactions or sensations up to the time when complete comfort is established.
7. The physiological reactions and sensations of a person leaving a cooled space and re-entering the hot outside, and the duration of these reactions.

This report deals with the results of the continued study during the summer of 1936 and gives extensive facts concerning items 1 to 4 above, and some added information concerning items 5 to 7.

RELATION BETWEEN THE DESIRED EFFECTIVE TEMPERATURE FOR SUMMER COMFORT AIR CONDITIONING AND THE WEATHER CONDITIONS OF DIFFERENT GEOGRAPHICAL LOCATIONS

In order to indicate the degree to which different geographical locations with different weather conditions affect the indoor cooling requirements for summer air conditioning, three distinct studies were carried on during the past summer, by or in cooperation with the Research Laboratory:

1. A brief study was carried on by the Ontario Research Foundation, Toronto, Ontario.
2. A similar study was carried on by the Engineering Experiment Station of the Agricultural and Mechanical College of Texas, College Station, Texas.
3. This and other phases of the subject outlined above were studied by the Research Laboratory of the Society in Pittsburgh.

Logs of the pertinent weather conditions in Toronto and Pittsburgh for the periods of the studies were made. (See Figs. 14 to 16).

The Toronto study was made during July, and the Texas study during July and August. The Pittsburgh study as regards this phase of the subject was made at intervals during the latter part of June, July, August and the early part of September. Included in the results analyzed in this phase of the subject are data from the similar study made and reported last year.

All three studies were carried on under approximately the same laboratory conditions as pertained in the 1935 study in Pittsburgh. In each case, an inside, well-insulated room was equipped so that it could be cooled, with provisions for controlling at will the atmospheric conditions. Tests of two general types were made in each instance. In one type of test the atmospheric conditions in the test room were made to vary slowly over a period of time along a pre-

determined course when plotted on the psychrometric chart, in order to traverse the probable comfort zone, or some portion thereof. The usual practice was to bring the subjects to equilibrium with a predetermined constant condition within the test room, after which the condition was made to vary slowly, while the reactions of the subjects were recorded at frequent intervals, according to the following scale of comfort which has been used in earlier laboratory reports:

"4"—comfortable.
"3"—comfortably cool, *i.e.*, that the subject was not particularly uncomfortable, but would for the sake of comfort desire a warmer rather than a cooler condition.
"2"—cool.
"1"—cold.
"5"—comfortably warm.
"6"—too warm.
"7"—hot.

At the same time, the degree of sensible perspiration was observed according to the following scale:

"0"—forehead or body, dry.
"1"—forehead or body, clammy.
"2"—forehead or body, damp (perspiration just visible).
"3"—forehead or body, wet (sweat covering the surface, frequently in drops).
"4"—perspiration on the forehead runs down, or perspiration on the body runs down or wets through clothing.

In the second type of test, the subject after a brief walk out-of-doors recorded his feeling of comfort, degree of perspiration, body temperature and other reactions before entering the conditioned room, and again immediately upon entering, and at frequent intervals thereafter, while the temperature and humidity of the atmosphere in the room were kept constant.

These subjects participating in this phase of the study were normal, healthy, college students, ranging in age from 19 to 25 years. Subjects were only accepted for the study after undergoing a medical examination, in order to eliminate those who might be abnormal in any way, particularly as regards their temperature regulatory reactions.

The comfort reactions of the subjects in the tests made at different effective temperatures in Toronto are given in Fig. 1. The feeling of warmth observed in the three studies, and plotted against effective temperature in Figs. 1 to 4, show similar characteristics. In all cases the numerical index of the feeling of warmth increases with increase in effective temperature, and by inspection, the range of effective temperature over which a comfort feeling of "4" predominates may be picked out. To make this comparison easier, the percentages which the expressions of each degree of feeling of warmth are of the total number of expressions for any effective temperature are tabulated at the top of the charts. In the Toronto study, it was soon observed that on many days the outside temperature, particularly in the morning, was lower than that desired for comfort indoors. Since different reactions may be expected on cold and warm days, they are separated so that tests made on warm days are plotted in Fig. 1, and tests made on cold days are plotted in Fig. 2. The data plotted in Fig. 1, for those days when the desired inside

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temperature was below the outside temperature at the time of entering the test, show a comfort feeling of "4" predominating over the effective temperature range from 67.5 to 71.5 deg. The maximum percentage of observations indicating a comfort rating of "4" actually occurred at 70 deg effective temperature.

The data plotted in Fig. 2 are for observations on days when the inside

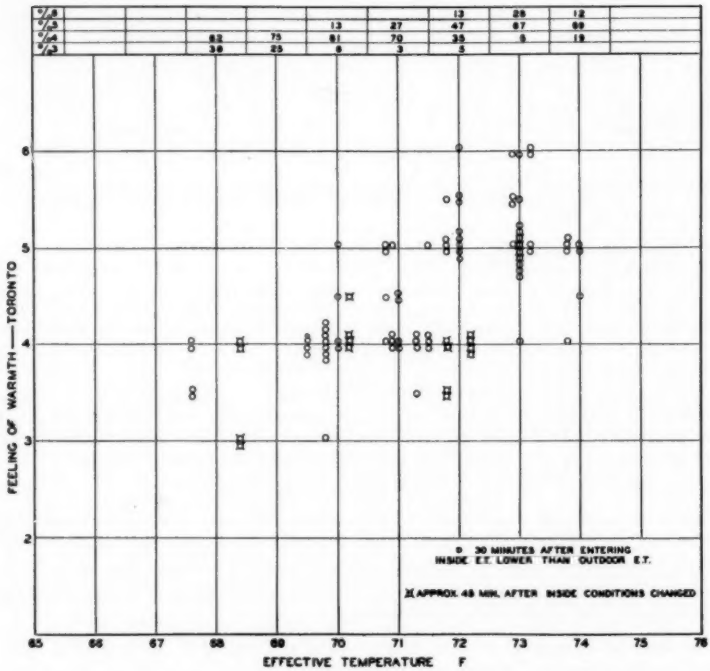


FIG. 1. RELATION BETWEEN EFFECTIVE TEMPERATURE AND FEELING OF WARMTH IN TORONTO ON DAYS WHEN OUTSIDE TEMPERATURE WAS WARMER THAN CONDITIONED TEST ROOM. PERCENTAGE OF EACH NUMERICAL INDEX OF COMFORT IS GIVEN IN UPPER PART OF CHART

conditions were warmer than the outside conditions at the time of entering the tests. Also plotted in this chart are observations on subjects entering the tests without coats. For the subjects entering the tests with coats the maximum percentage of comfort occurred at about 69 deg effective temperature, while for subjects entering without coats (see percentages in parentheses), the highest percentage of comfort was found at from 71 to 73 deg, indicating that comfort is realized at from 2 to 3 deg effective temperature higher

when no coat is worn. This will be discussed further in a later part of the paper.

The results of tests in Texas, given in Fig. 3, show a maximum percentage of 88 per cent of the subjects indicating comfort at 73 deg effective temperature, while over 50 per cent of the reactions were comfortable over an effective temperature range from 69.5 to 74.5 deg. The results of tests in Pittsburgh, given in Fig. 4, show maximum percentage indicating comfort at

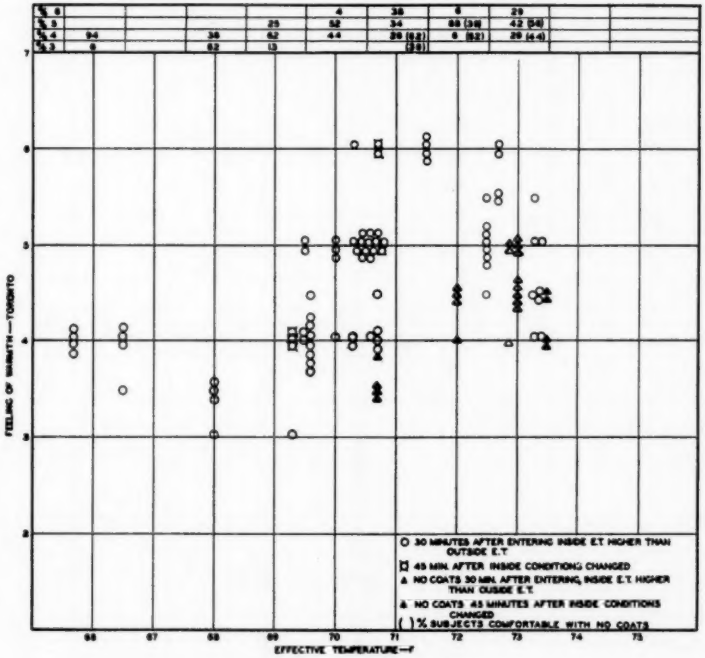

FEELING OF WARMTH — TORONTO
EFFECTIVE TEMPERATURE — F
O 30 MINUTES AFTER ENTERING INSIDE E.T. HIGHER THAN OUTSIDE E.T.
45 MIN. AFTER INSIDE CONDITIONS CHANGED
NO COATS 30 MIN. AFTER ENTERING, INSIDE E.T. HIGHER THAN OUSIDE E.T.
NO COATS 45 MINUTES AFTER INSIDE CONDITIONS CHANGED
( )% SUBJECTS COMFORTABLE WITH NO COATS

FIG. 2. RELATION BETWEEN EFFECTIVE TEMPERATURE AND FEELING OF WARMTH IN TORONTO ON DAYS WHEN OUTSIDE TEMPERATURE WAS COOLER THAN CONDITIONED TEST ROOM. PERCENTAGE OF EACH NUMERICAL INDEX OF COMFORT IS GIVEN IN UPPER PART OF CHART

between 72 and 73 deg effective temperature, with more than 50 per cent of the subjects indicating comfort between 68 and 75 deg.

In order to compare the reactions of the subjects more easily, their comfort ratings from "2" to "6" are plotted against effective temperature for the Pittsburgh study in Fig. 5. The percentages giving a comfort rating of "4," or ideal comfort, are plotted for each study in Fig. 6. While there is considerable variation between the three curves indicating a comfort feeling of "4"

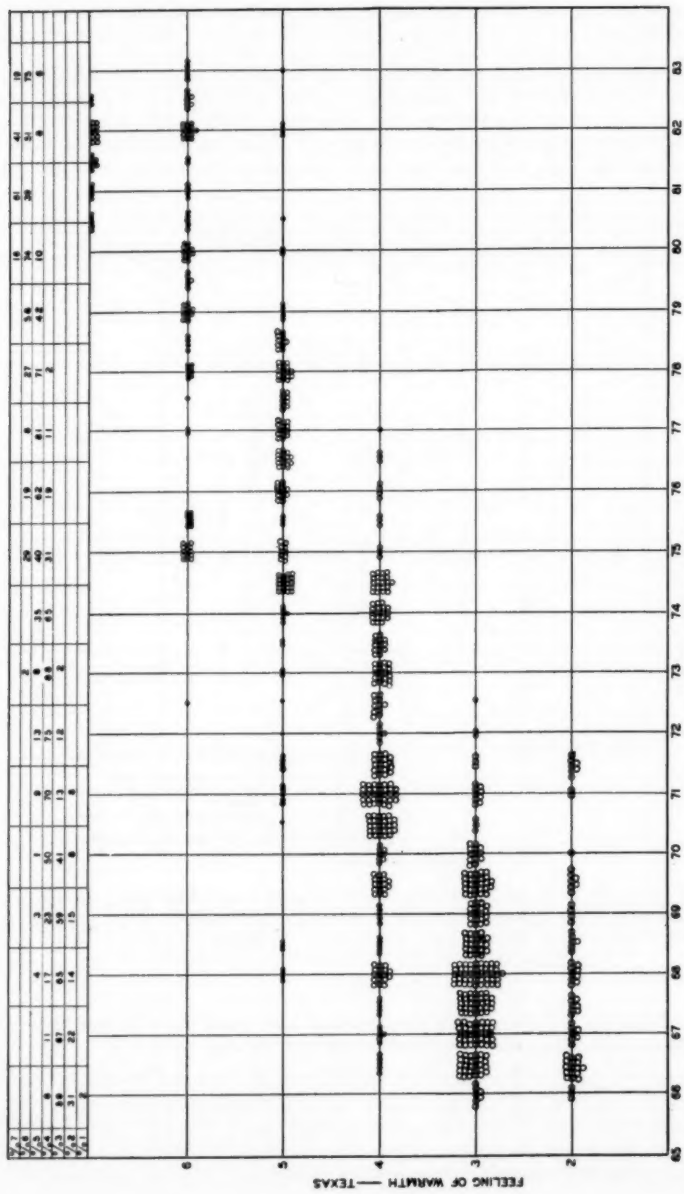


FIG. 3. RELATION BETWEEN EFFECTIVE TEMPERATURE AND FEELING OF WARMTH FOUND IN TEXAS STUDY. PERCENTAGE OF EACH NUMERICAL INDEX OF COMFORT IS GIVEN IN UPPER PART OF CHART

in the three phases of the study, particular significance is attached to the fact that maximum comfort was indicated in the Toronto study at approximately 70 deg effective temperature, while in the Pittsburgh and Texas studies maximum comfort was indicated at 72.5 and 73.0 deg effective temperature, respectively. The one-half degree variation in maximum comfort between the

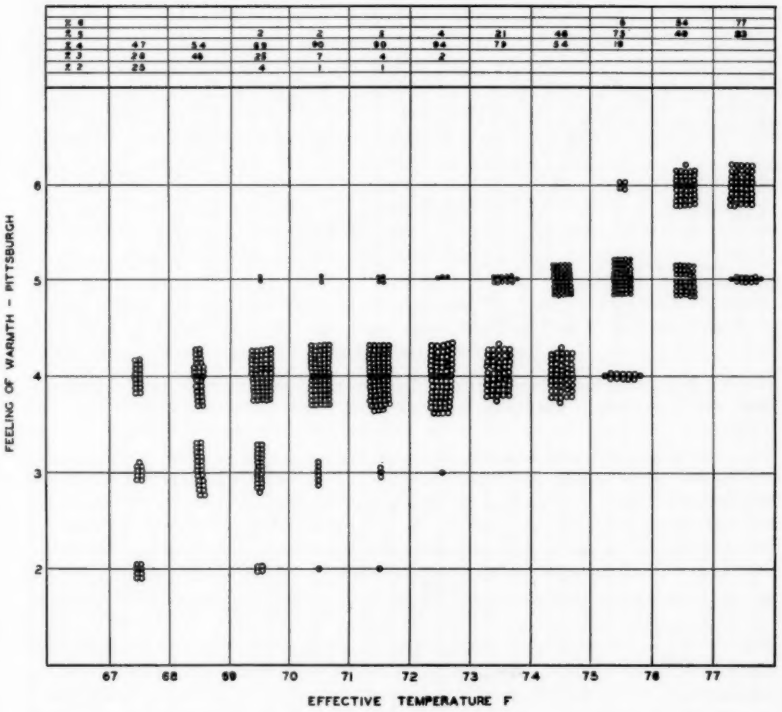


FIG. 4. RELATION BETWEEN EFFECTIVE TEMPERATURE AND FEELING OF WARMTH IN PITTSBURGH STUDY. PERCENTAGE OF EACH NUMERICAL INDEX OF COMFORT IS GIVEN IN UPPER PART OF CHART

latter two studies is not considered to be significant, and for all practical purposes they may be considered as the same. Some significance may, however, be attached to the fact that the lower limit of comfort in the Pittsburgh study was at a lower effective temperature than that for the Texas study. On the other hand, this difference of approximately 1½ deg may result from the greater number of subjects in the Pittsburgh study. For comparison, a similar curve resulting from a study of the comfort zone in Boston

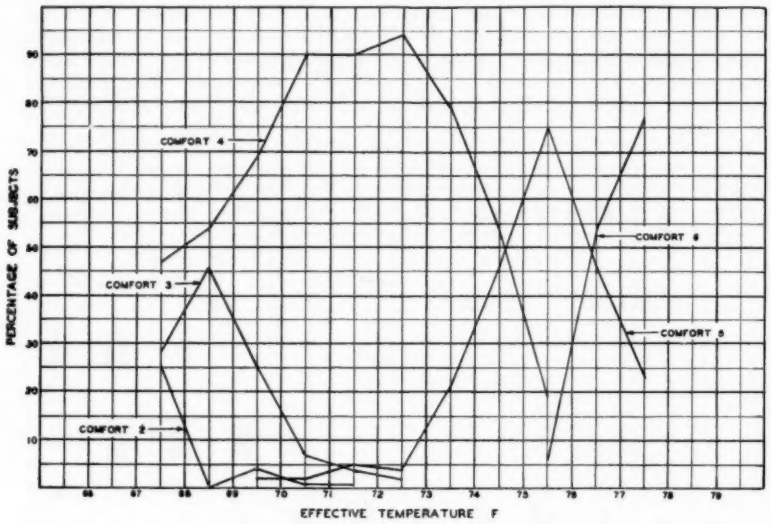


FIG. 5. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OF SUBJECTS INDICATING DIFFERENT FEELINGS OF WARMTH FOR ALL TESTS MADE IN PITTSBURGH

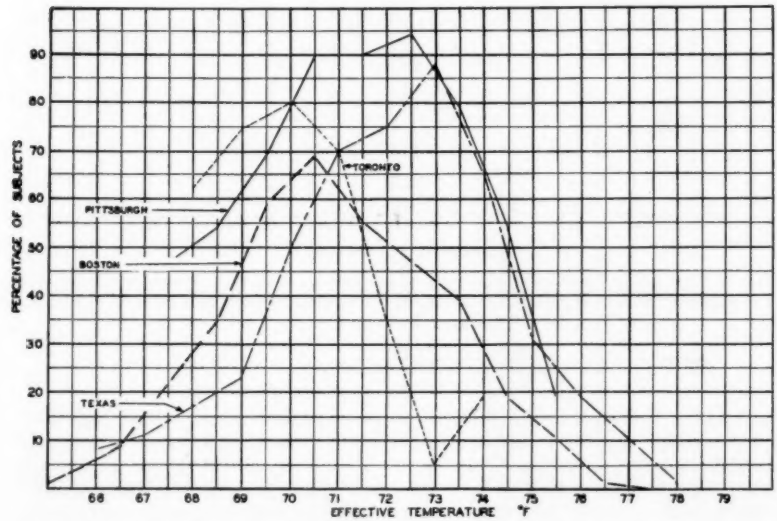


FIG. 6. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OF SUBJECTS INDICATING A COMFORT FEELING OF 4 IN TORONTO, PITTSBURGH, TEXAS AND BOSTON

by Professor Yaglou [2] in July, August and September of 1927 is also plotted. The close relation between this curve and that resulting for Toronto in this study may be due to the similarity of the summer climate of the two cities as regards their average summer temperature.

From the three curves giving a comfort feeling of "4," a comfort zone for each of the three districts may be established if agreement may be had on the percentage of persons who should be expected to be ideally comfortable in a conditioned space. If 70 per cent of all persons occupying a conditioned space should be comfortable, then the Toronto study indicates a comfort zone from a little below 68.5 to 71 deg effective temperature. The Pittsburgh study indicates a zone ranging from 69.5 to almost 74 deg, while the Texas study indicates a range from 71 to 74 deg. Keeping in mind the fact that maintaining a higher temperature results in greater economy in cooling, and also the fact that with higher effective temperatures the cold shock upon entering a conditioned space is diminished, it seems desirable to recommend for general use temperatures nearer the upper limits rather than the lower limits of these ranges. This would seem to dictate an effective temperature of 70 or 71 deg for Toronto, and about 73 deg for Pittsburgh and Texas. This would also indicate that all localities having a summer condition as warm or warmer than Pittsburgh should use summer cooling effective temperatures of about 73 deg, while for cooler climates with lower mean summer temperatures, lower effective temperatures should be maintained.

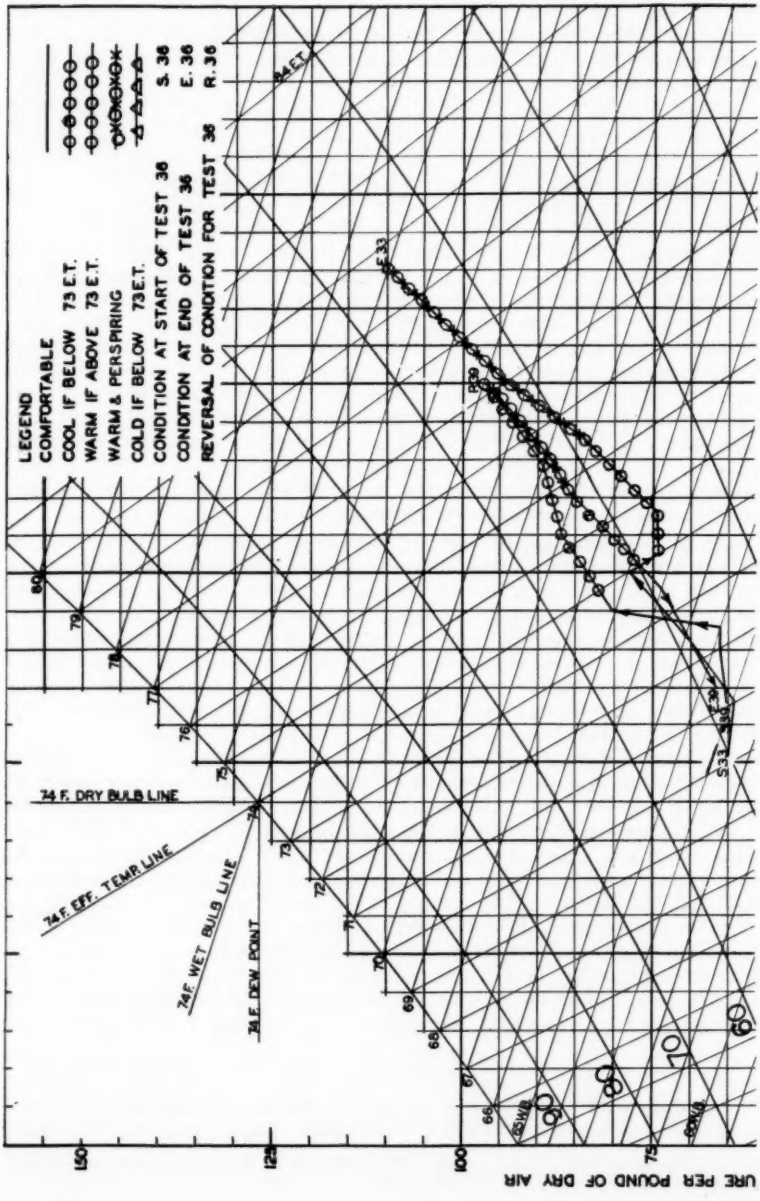
The studies here reported and the Pittsburgh study of last year indicate that no variation in the desired indoor cooling temperature accompanies a daily or even weekly change in outside weather, excepting possibly on days when the outside is cooler than the inside. It appears, rather, that the desired indoor temperature depends upon the average outside temperature during the hot season. Insofar as these deductions are true, it should be a simple matter to base the requirement for any locality on a study of its average summer temperature in relation to that of Pittsburgh and Toronto. Since the difference in the requirements of these two localities is only 2 or 3 deg, and since Toronto seems to have about as low an average temperature as any important city on the continent, possible errors in such an interpolation or extrapolation should not be great. A study of these variations is being made.

Fig. 7 shows the results of a few tests made in Toronto when the conditions within the test room were made to vary so as to follow a predetermined course when plotted on the psychrometric chart. The degree of comfort, as indicated by symbols on the curves, shows that comfort was observed for effective temperatures above 67.5 deg, agreeing fairly well with the results plotted in Fig. 6. These tests, however, indicate a higher maximum temperature for comfort than the tests plotted in Fig. 6. They also indicate a sensible perspiration line at about 76 deg effective temperature, which is in agreement with the work in the earlier study made in Pittsburgh.

VARIATION IN RELATIVE HUMIDITY AT A GIVEN EFFECTIVE TEMPERATURE

The study made in Pittsburgh during the summer of 1935 included a few tests which indicated that over a considerable range, variations in relative

[2] The Summer Comfort Zone: Climate and Clothing, by C. P. Yaglou and Philip Drinker. (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 269).


LEGEND
COMFORTABLE
COOL IF BELOW 73 E.T.
WARM IF ABOVE 73 E.T.
WARM & PERSPIRING
COLD IF BELOW 73 E.T.
CONDITION AT START OF TEST 36 S. 36
CONDITION AT END OF TEST 36 E. 36
REVERSAL OF CONDITION FOR TEST 36 R. 36
74 F. DRY BULB LINE
74 F. EFF. TEMP. LINE
74F WET BULB LINE
74 F. DEW POINT
84 E.T.
URE PER POUND OF DRY AIR

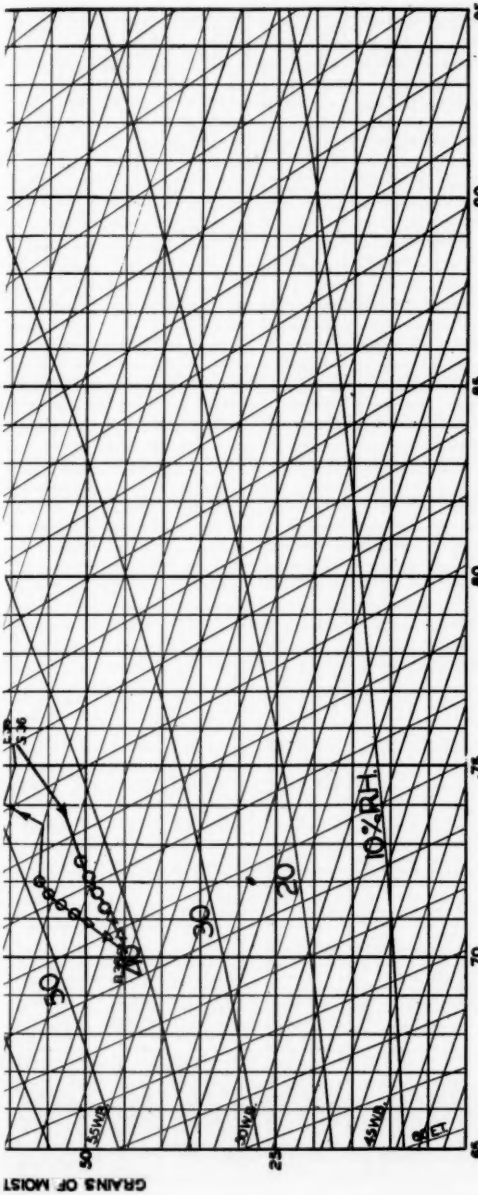


FIG. 7. VARIATION IN ATMOSPHERIC CONDITIONS MAINTAINED DURING THREE TESTS IN TORONTO AND THE REACTIONS OF THE SUBJECTS TO THESE CONDITIONS

TABLE 1—RESULTS OF 52 TESTS IN 73 DEG EFFECTIVE TEMPERATURE AND VARIOUS RELATIVE HUMIDITIES IN PITTSBURGH AFTER EXPOSURE TO HOT ATMOSPHERES EITHER CONTROLLED IN A *Hot Room* OR ENCOUNTERED WHILE WALKING OUTSIDE

TEST NO.	DATE	TEST ROOM E.T.	TEST ROOM R.H.	ENTERED FROM	E.T.	R.H.	FEELING OF WARMTH B.E.	A.E.	10 MIN.	20 MIN.	30 MIN.	40 MIN.	50 MIN.	60 MIN.	DEGREE OF SENSIBLE BODY PERSPIRATION A	B	C	AVE.	BODY PERSPIRATION ENDS MIN. A	B	C	AVE.	DEGREE OF SENSIBLE FOREHEAD PERSPIRATION A	B	C	AVE.	FOREHEAD PERSPIRATION ENDS MIN. A	B	C	AVE.
30	7/7/36	75	50	HOT ROOM	87	65	7	3.0	3.7	4.0	4.0	4.0	4.0	4.0	3.0	3	3	3	14	6	22	14	2	2	3	2.3	10	10	10	10
33	7/8/36						7	2.5	3.5	3.5	4.0	4.0	4.0	4.0	—	3	3	3	—	10	30	20	—	3	3	3	—	6	6	6
34	7/8/36						7	3.0	3.0	3.5	4.0	4.0	4.0	4.0	—	2	2	2	—	20	30	25	—	2	2	2	—	8	8	8
66	7/20/36						7	2.7	4.3	4.3	4.3	4.0	4.0	4.0	4	4	4	4	15	12	20	15.7	3	3	3	3	11	20	10	14
58	7/21/36						7	2.7	4.0	4.0	4.0	4.0	4.0	4.0	3	4	4	3.7	15	20	40	25	3	3	3	3	10	12	12	11
AVE. FROM HOT ROOM							7	2.8	3.7	3.9	4.1	4.0	4.0	4.0	3.3	3.2	3.2	3.1	14.6	13.6	28.4	19.9	2.7	2.6	2.8	2.6	10	11	9	10
31	7/7/36	75	50	OUTSIDE	76	39	6.0	3.0	4	4.0	4.0	4.0	4.0	4.0	3	—	1	2.0	40	—	30	35	1	—	1	1	40	—	6	23
32	7/7/36				77.6	31.5	6.7	4.3	4	4.0	4.0	4.0	4.0	4.0	1	1	2	1.3	10	6	10	8.7	1	1	2	1.3	4	6	6	5
35	7/9/36				79	48	6.0	5.0	3.5	4.0	4.0	4.0	4.0	4.0	—	1	2	1.5	—	8	10	9	—	1	1	1	—	6	4	5
56	7/20/36				76	52	5.3	4.0	4.3	4.7	4.3	4.0	4.0	4.0	1	1	1	1.0	15	10	20	15	1	1	2	1.4	16	10	8	11
57	7/20/36				78.8	32.5	6.3	4.0	4.3	4.3	4	4.0	4.0	4.0	2	0	1	1.0	10	0	15	8.3	2	—	1	1	8	—	4	4
59	7/21/36				75.8	41	5.8	4.6	4.5	4.0	4	4.0	4.0	4.0	1	—	0	.5	11	—	0	5.5	1	—	0	.6	4	—	0	2
60	7/21/36				76	36	6.0	3.0	4.0	4.0	4	4.0	4.0	4.0	2	—	2	2.0	6	—	10	8	2	—	1	1.5	10	—	6	8
AVE. FROM OUTSIDE							5.8	3.7	4.0	4.1	4.1	4.0	4.1	4.0	1.6	0.75	1.3	1.3	15.3	6	13.6	12.8	1.3	1	1.1	1.1	14	7	5	6
AVE. ALL TESTS 73 E.T. 30% R.H.							6.4	5.25	3.85	4.0	4.1	4.0	4.05	4.0	2.45	2.0	2.25	2.2	15	7.1	21	14.35	2.0	1.8	2	1.8	12	9	7	8
26	7/6/36	75	50	HOT ROOM	87	62	7	3.5	3.7	3.7	3.7	3.7	4.0	4.0	3	3	3	3	25	10	20	18	2	2	3	2.3	20	6	10	12
27	7/6/36						7	3.0	4.0	4.0	4.0	4.0	4.0	4.0	2	2	3	2.3	15	10	8	10	2	2	3	2.3	20	10	10	13
36	7/9/36				7	2.7	3.0	4.0	4.0	4.0	4.0	4.0	4.0	3	3	3	3	40	14	35	30	1	3	3	2.3	6	8	15	10
37	7/9/36				6.5	2.5	4.0	4.0	3.5	3.5	3.5	3.5	3.5	3	—	2	2.5	2.0	—	20	25	2	—	2	2	15	—	8	12
AVE. FROM HOT ROOM							6.7	2.9	3.7	3.9	3.9	3.8	3.9	3.9	2.8	2.7	2.8	2.7	27.5	10.7	20.7	20.7	1.7	2.3	2.8	2.2	16	8	11	12
28	7/6/36	75	50	OUTSIDE	74.7	33	5.6	3.5	4.5	4.5	4.0	4.0	4.0	4.0	2	—	1	1.5	50	—	10	30	2	—	1	1.5	50	—	10	30
29	7/6/36				75.2	35	6.0	3.3	4.7	4.3	4.0	4.0	4.0	4.0	2	1	1	1.3	30	8	4	14	2	1	1	1.3	16	10	10	12
38	7/9/36				84.6	55	7.0	2.3	3.7	4.3	4.0	4.0	4.0	4.0	3	4	2	3	40	30	10	27	2	3	2	2.3	15	8	10	11
AVE. FROM OUTSIDE							6.1	3.4	4.3	4.4	4.3	4.2	4.2	4.2	2.7	2.5	1.3	1.9	40	19	8	23.7	2	2	1.3	1.7	27	9	10	18
AVE. ALL 73 E.T. 50% R.H.							6.6	3.1	4.3	4.3	4.1	4.0	4.05	4.05	2.7	2.6	2.1	2.3	33.5	14.8	14.4	22.2	1.9	2.2	2.1	2.0	21	9	11	15
5	6/24/36	75	60	HR	87	60	7.0	2.3	3.6	3.6	4	4	4	4	3	2	2	2.3	20	10	30	20	1	2	3	2	10	10	20	13
14	6/26/36			OS	71.3	58	5.0	4.3	4.3	4.0	4	4	4	4	2	1	0	1	10	10	0	7	1	1	1	1	10	10	10	10
AVE. ALL 73 E.T. 60% R.H.							6.2	3.3	3.9	3.8	4	4	4	4	2.5	1.5	1	1.7	15	10	15	14	1	1.5	2	1.5	10	10	15	11.5
6	6/24/36	75	70	HOT ROOM	87	60	7	2.3	3.7	3.7	3.7	3.7	4.0	4	4	3	2	3	20	20	20	20	1	2	2	1.7	10	10	10	10
23	7/8/36						7	2.7	3.3	3.7	3.7	4	4.0	4	3	2	2	2.3	30	10	30	23	2	2	2	2	10	10	8	9
24	7/8/36						7	2.3	3.3	3.5	3.7	3.7	3.7	3.7	4	3	2	3	20	20	60	33	2	3	2	2.3	10	20	20	17
39	7/10/36						7	2.3	3.3	3.3	3.3	3.7	3.7	3.7	3	3	3	3	40	20	60	43	2	3	3	2.7	10	20	2	17
AVE. FROM HOT ROOM							7	2.4	3.4	3.5	3.6	3.8	3.9	3.9	3.5	2.9	2.5	2.8	28	22	40	27	1.8	2.5	2.3	2.2	10	15	15	13

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humidity did not affect the feeling of warmth, provided the proper effective temperature was maintained. This phase of the study was not, however, conclusive and the inference was the subject of considerable debate among those discussing the subject. Accordingly, a large part of the work in Pittsburgh during the past summer was directed toward clearing up this point. A series of 52 tests, all at 73 deg effective temperature, and with variations in relative humidity from 30 to 90 per cent, was made, and the results are presented in Table 1.

The most important effect of the moisture content of the air is its reaction on the feeling of warmth; however, it is recognized that it may have other important, though possibly more subtle, consequences. In this study, importance was placed primarily on its effect on the feeling of warmth. An attempt was made, however, to obtain the opinions of those participating in the test regarding other reactions experienced, as well as their general impression of conditions.

The air conditioning equipment in the Laboratory at Pittsburgh is not well adapted to maintain relative humidities above 80 per cent at dry-bulb temperatures above 70 deg during the summer, and in order to obtain these conditions it was necessary to augment the regular air conditioning system by admitting steam and spraying water into the test room. These applications frequently resulted in *steam* and other odors and the appearance of fog in some localized parts of the room. Probably as a consequence of these disturbances, the subjects frequently indicated their displeasure with the high humidities supplied, although they admitted it had nothing to do with their feeling of warmth. This phase of the subject should be given further study.

In order to eliminate the effect of the variable outside temperature, about one-half of the tests were made with the subjects entering the test room from a nearby, artificially heated room, instead of from the outside. In these tests the hot room was always maintained at 87 deg effective temperature with from 50 to 60 per cent relative humidity. The subjects remained in this hot room approximately 15 min, or in the case of each individual subject, until his feeling of warmth reached a comfort index of "7." This had the effect of giving a series of tests in which the variable effect of outside weather was eliminated. In all cases the subject entered the test room immediately after recording his feeling of warmth, degree of perspiration, and body temperature. The feeling of warmth and degree of perspiration were again recorded immediately after entering the test room and at frequent intervals thereafter during the period of the test, which lasted until all of the subjects reached a condition of comfort. Table 1 gives the feeling of warmth and the degree of perspiration on the body before entering, the feeling of warmth or the cold shock immediately after entering, the feeling of warmth at intervals after entering, and the time required for the disappearance of sensible perspiration from the forehead and from the body.

The relation between the forehead perspiration before entering the test and the time required for perspiration to disappear after entering the test is plotted in Fig. 8. Different symbols and curves give the relationship for tests in which the cooled room was entered from another hot room and for tests in which it was entered from the outside. While there is considerable variation in the test points for individual subjects in different tests, it is quite

apparent that the time required for perspiration to disappear increases with the increased degree of perspiration before entering. In each case there is a slight increase in the time required for perspiration to disappear, with the same degree of perspiration before entering, when the subject entered from the outside over that required when the subjects entered from the hot room. This may be accounted for by the fact that the hot room was usually at a higher temperature than the outside, so that the subjects reached the high degree of perspiration indicated in a shorter period of time and with less rise in body temperature.

The curves for the five different relative humidities show no measurable progressive change from lower to higher moisture content of the air. The curves as drawn from the data show that 10.5, 16.5, 16, 15 and 14 min were required for sensible perspiration to disappear in atmospheres of 73 deg effective temperature and 30, 50, 70, 80 and 90 per cent relative humidity, respectively, when a perspiration rate of "2" pertained before entering the test room from the outside. Similarly, 8, 10.5, 10.5, 10 and 8.5 min were required for the disappearance of sensible perspiration from the subjects in atmospheres of 73 deg effective temperature and 30, 50, 70, 80 and 90 per cent relative humidity, respectively, after entering the test room from the hot room with a sensible perspiration rate of "2." Observations plotted in Fig. 8 show that in a number of cases perspiration lingered longer in atmospheres of 80 and 90 per cent relative humidity than in lower humidities. While there is some indication of a slightly increased length of time for the disappearance of perspiration in high relative humidities, the increase is less than the errors in measuring the effect or the variations resulting from different individuals at different times. The variations may, therefore, be assumed to be of no significance. This is contrary to what might logically be expected from general experience in the rates of drying of various materials in air of the same temperature but with different relative humidities. It must be kept in mind, however, that the disappearance of sensible perspiration is more than an ordinary drying process. The disappearance of perspiration must be greatly affected by the time necessary for the body to cease exuding perspiration after entering an atmosphere of lower temperature. Also, the evaporation of perspiration from the body of a person is affected both by the temperature and moisture content of the air and the temperature of the body itself. Since the surface temperature of the body, even when under the greater cooling rate met with in the cooled atmosphere, does not fall greatly, the vapor pressure of the moisture on the surface is considerably higher than the vapor pressure of the atmosphere at 73 deg effective temperature with either high or low relative humidity. From this point of view, the apparent insignificance of the change in disappearance of perspiration with change in relative humidity becomes less surprising. Further study of this phase of the subject is recommended.

One should expect to find a significant correlation between the degree of perspiration before entering a cooled atmosphere and the time required to reach comfort after entering. A similar correlation should be expected between the feeling of warmth before entering a cooled atmosphere and the time required to reach comfort after entering. However, plotting these relationships from the data in Table 1 shows no significant correlation.

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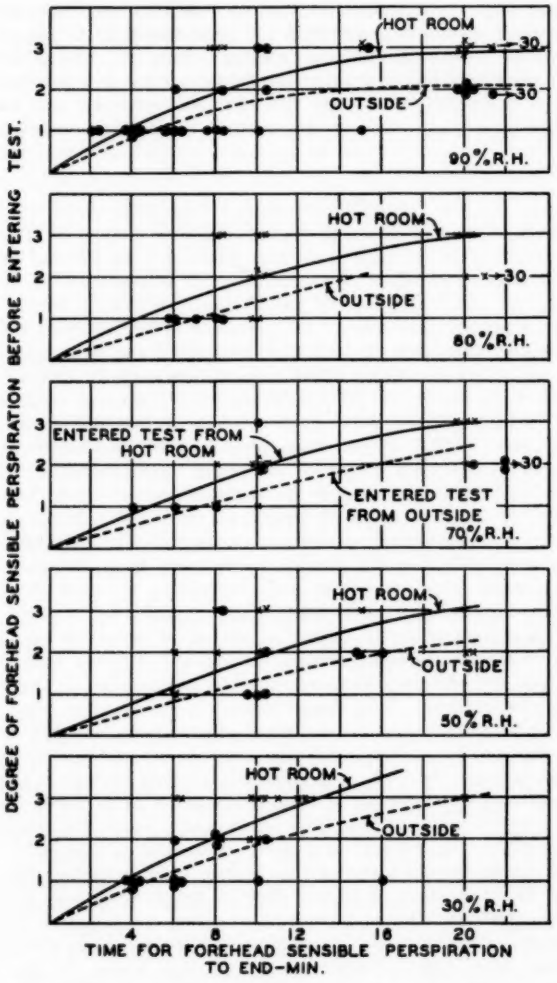

FIG. 8. RELATION BETWEEN DEGREE OF SENSIBLE PERSPIRATION ON FOREHEADS OF SUBJECTS BEFORE ENTERING TESTS AND TIME REQUIRED FOR SENSIBLE PERSPIRATION TO DISAPPEAR IN ATMOSPHERES OF 73 DEG ET AND VARIOUS RELATIVE HUMIDITIES

Symbols (x) and solid line curves for tests following exposure to 87 deg E T in a hot room; circles (o) and broken line curves for tests following exposure to hot outdoor conditions.

The relation between the cold shock or the feeling of discomfort immediately after entering a cooled atmosphere, at 73 deg effective temperature with various relative humidities, and the degree of perspiration on the body before entering on the one hand, and the feeling of warmth before entering the conditioned space on the other, are shown in Figs. 9 and 10. There is a reasonably good correlation in both cases, showing that the cold shock upon entering a cooled space increases with the degree of warmth and the degree of perspira-

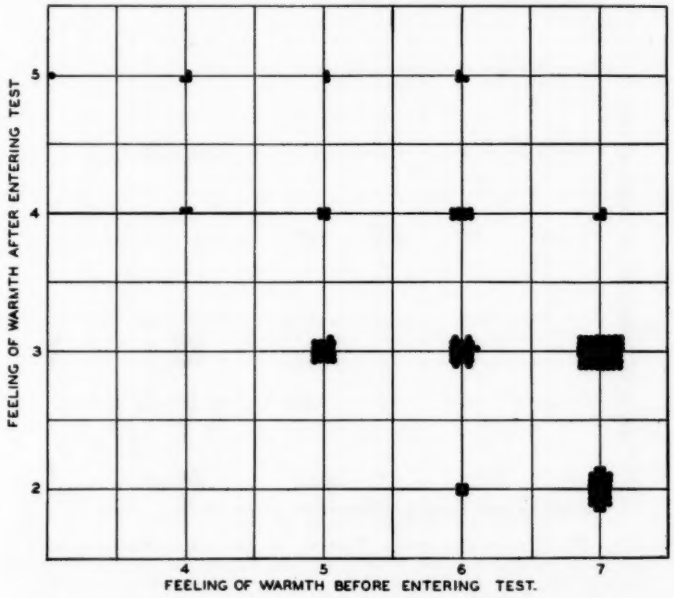

FIG. 9. RELATION BETWEEN FEELING OF WARMTH BEFORE ENTERING TEST AND SHOCK OR FEELING OF COLD IMMEDIATELY AFTER ENTERING AN ATMOSPHERIC CONDITION OF 73 DEG ET AND VARIOUS RELATIVE HUMIDITIES

tion experienced before entering. As the feeling of warmth before entering increases from "4" to "7," the feeling of warmth or the cold shock immediately after entering decreases from approximately "4" to "2.5," and, while the degree of perspiration before entering increases from "0" to "4," the cold shock after entering decreases from "4" to "2.5." There is no noticeable variation in these relationships for tests made at different relative humidities and 73 deg effective temperature, proving rather conclusively that the cold shock upon entering a cooled space is independent of the relative humidity at a given effective temperature.

The relation between time required for sensible perspiration to disappear from the body and the time required to reach comfort after entering a cooled

atmosphere at 73 deg effective temperature and various relative humidities is shown in Fig. 11. The data do not indicate any definite curve or relationship; however, the fact that few test points are found in the lower right hand portion of the chart, particularly to the right and below a line through the "10 min comfort—zero perspiration" and the "50 min comfort—40 min perspiration" points, indicates that comfort is almost always reached not later than 10 min after the disappearance of body perspiration, and that it may be reached earlier.

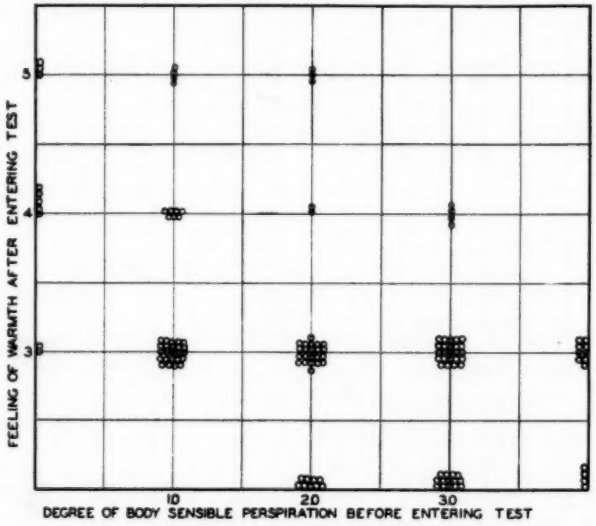

FIG. 10. RELATION BETWEEN FEELING OF SENSIBLE PERSPIRATION ON BODY BEFORE ENTERING TEST AND SHOCK OF FEELING OF COLD IMMEDIATELY AFTER ENTERING AN ATMOSPHERIC CONDITION OF 73 DEG ET AND VARIOUS RELATIVE HUMIDITIES

This involves a fine point in consideration of comfort. Strictly speaking, it may be assumed that no one is comfortable while still moist with perspiration.

Influence of Perspiration

However, in these tests, comfort based upon a feeling of warmth only was more strongly emphasized. Instances where perspiration lingered after a comfortable feeling of warmth was registered probably include only cases where considerable perspiration was held on the body and in the clothing before entering the test, which evaporated rather slowly and hence did not greatly affect the feeling of warmth. As a corollary to the above assumption, it may likewise be stated that perspiration rarely disappeared earlier than 10 min before a comfortable feeling of warmth was established.

The relation between the length of time, after entering the conditioned space, required to establish comfort and the time for sensible perspiration to disappear from the forehead is given in Fig. 12. No sharp relationship is noted but the general distribution of the points indicates the reverse of that brought out by Fig. 6. Few points are found in the upper left hand part of the chart

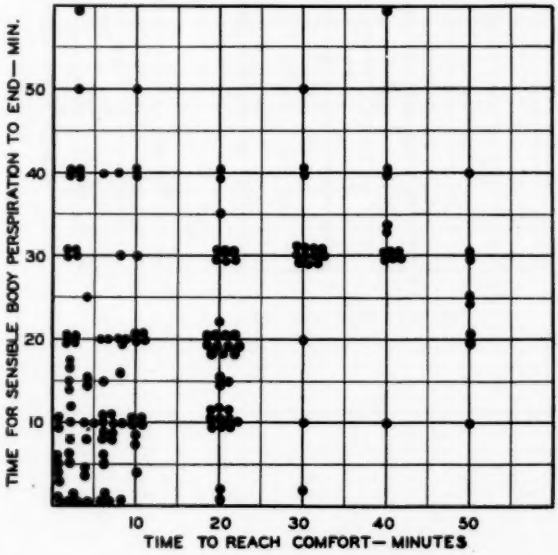

FIG. 11. RELATION BETWEEN TIME REQUIRED FOR SUBJECTS TO REACH COMFORT AND TIME FOR SENSIBLE BODY PERSPIRATION TO DISAPPEAR FOR ALL SUBJECTS ENTERING TESTS AT 73 DEG ET AND VARIOUS RELATIVE HUMIDITIES

showing that forehead perspiration usually disappears before comfort is established and rarely more than 10 min later.

Time Required to Reach Comfort

Rise in body temperature resulting from exposure to, and exercise in, hot atmospheres may be assumed to have a material effect on the time required to reach comfort. This relationship is shown in Fig. 13. While the test points show no close relationship, there is an apparent tendency for the time required to reach comfort to increase with the degree of body temperature above normal at the time of entering the test.

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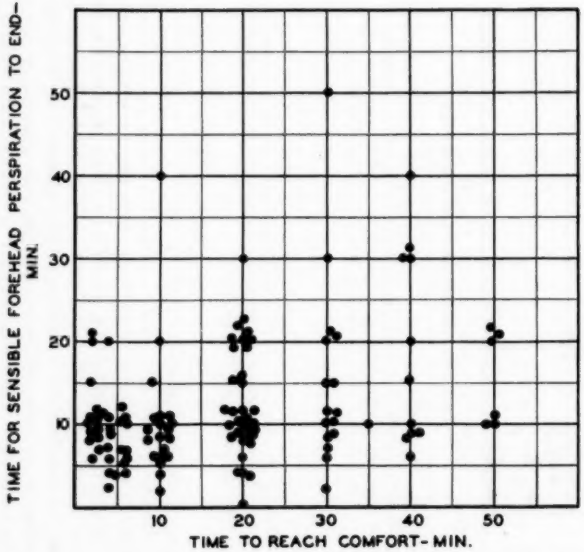

FIG. 12. RELATION BETWEEN TIME REQUIRED FOR SUBJECTS TO REACH COMFORT AND TIME FOR SENSIBLE FOREHEAD PERSPIRATION TO DISAPPEAR FOR ALL SUBJECTS ENTERING TESTS AT 73 DEG ET AND VARIOUS RELATIVE HUMIDITIES

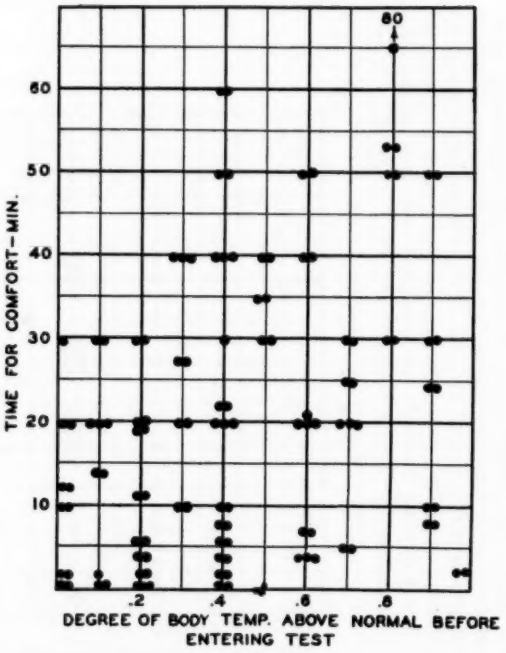

FIG. 13. RELATION BETWEEN ELEVATION OF BODY TEMPERATURE ABOVE NORMAL BEFORE ENTERING TEST TO TIME REQUIRED FOR THE ESTABLISHMENT OF COMFORT AFTER ENTERING ATMOSPHERES OF 73 DEG ET AND VARIOUS RELATIVE HUMIDITIES

RELATION BETWEEN THE AGE, SEX AND HEALTH OF SUBJECTS, AND THE CLOTHING WORN BY THEM, AND THE REQUIRED EFFECTIVE TEMPERATURE FOR SUMMER COMFORT AIR CONDITIONING

A most important question concerning cooling requirements for summer comfort air conditioning is the possible variation in the desired indoor air temperature based upon the age, sex and health characteristics of the persons involved. A comprehensive study of this phase of the subject was planned, but only relatively few data have been made available, and further study is urgently

TABLE 2—REACTIONS OF MEN AND WOMEN BEFORE AND AFTER ENTERING VARIOUS CONDITIONS; THE REACTIONS DURING *Warm Weather* ON DAYS WHEN THE OUTSIDE EFFECTIVE TEMPERATURE WAS ABOVE THAT IN THE TEST ROOM, AND FOR *Cool Weather* ON DAYS WHEN THE OUTSIDE EFFECTIVE TEMPERATURE WAS BELOW THAT MAINTAINED IN THE TEST ROOM

TEST ROOM E.T.	TIME	MEN								WOMEN							
		COATS				NO COATS				WARM WEATHER				COOL WEATHER			
		No.	UNDER 40	No.	OVER 40	No.	UNDER 40	No.	OVER 40	No.	UNDER 40	No.	OVER 40	No.	UNDER 40	No.	OVER 40
70	BEFORE ENTRY	10	4.3	6	5.0	25	5.1	4	4.5					9	5.8*	7	5.8*
	AFTER ENTRY		3.5		3.6		2.4		3.5						3.3		3.3
	15-25 MIN.		3.9		3.7		3.3		3.3						4.0		4.0
	AFTER 25 MIN.		3.9		3.7		3.5		3.5						4.0		4.0
	OUTSIDE		6.0		5.0		5.2		5.5						6.0		5.5
73	BEFORE ENTRY	8	5.3	10	5.2	37	5.0	12	4.7	12	4.7			20	6.1*	12	6.0*
	AFTER ENTRY		3.0		3.4		3.0		2.9		3.5				4.2		4.1
	15-25 MIN.		4.1		4.0		4.0		3.7		4.5				4.2		4.3
	AFTER 25 MIN.		4.1		4.0		3.9		4.0		4.0				4.5		4.3
	OUTSIDE		5.7		4.9		5.3		5.1		4.7				5.9		5.6
75	BEFORE ENTRY	7	6.0	11	4.7	23	5.4	15	5.0								
	AFTER ENTRY		4.5		4.0		3.8		3.8								
	15-25 MIN.		5.0		4.6		4.0		4.3								
	AFTER 25 MIN.		5.0		4.6		4.0		4.0								
	OUTSIDE		5.0		4.7		5.4		5.1								
78	BEFORE ENTRY			7	5.3	5	5.0	4	5.0					12	6.0*		
	AFTER ENTRY				3.2		4.0		4.0						3.9		
	15-25 MIN.				5.5		4.6		4.3						4.8		
	AFTER 25 MIN.				3.5		4.8		4.3						4.9		
	OUTSIDE				6.0		5.4		5.5						5.9		
77	BEFORE ENTRY			2	5.0	6	6.0	6	5.2	4	5.0	4	5.5				
	AFTER ENTRY				5.0		3.7		3.6		4.0		3.5				
	15-25 MIN.				6.0		4.3		5.0		5.0		4.5				
	AFTER 25 MIN.				6.0		5.0		4.8		5.0		4.5				
	OUTSIDE				6.0		5.3		5.4		5.0		5.0				

* Subjects exposed to hot atmospheres in another room before entering tests.

needed. During the latter part of the summer, men and women of various walks in life and of different ages were induced to enter the conditioned rooms and give their opinions concerning their feeling of warmth and other reactions. The results of this survey are shown in Table 2. These tests indicate in general the same temperature requirement for comfort as indicated by the trained subjects.

Men, both under and over 40 years of age, wearing coats, showed an average degree of warmth a little below "4" at 70 deg effective temperature, approximately "4" at 73 deg, and too warm at higher effective temperatures. There is a slight, though not pronounced, expressed desire for a higher temperature by the men over 40 years of age. However, this is not very significant. It would be of considerable interest to have similar data on the

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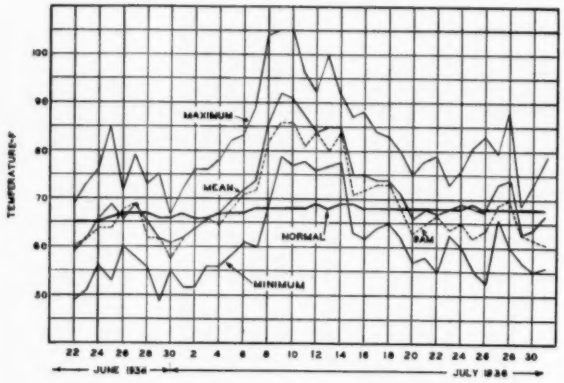
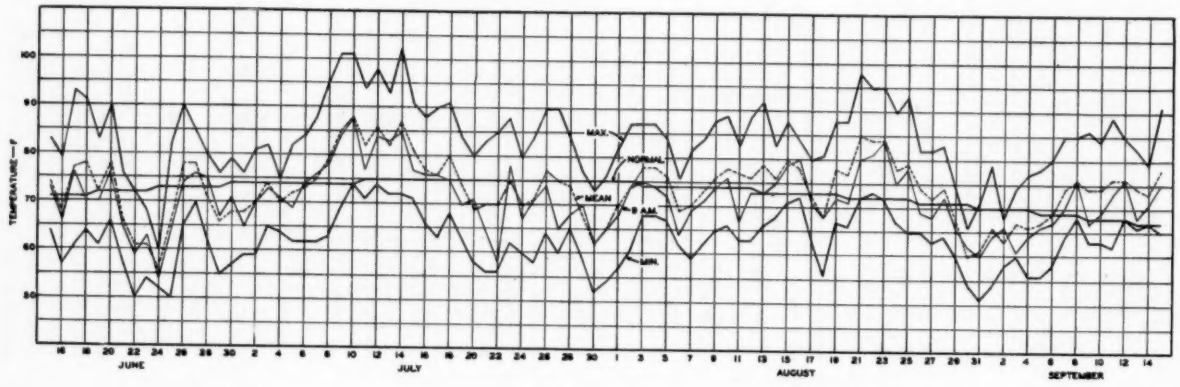


FIG. 14. (*Left*) LOG OF WEATHER CONDITIONS IN TORONTO JUNE 22 TO JULY 31, 1936

FIG. 15. (*Below*) LOG OF WEATHER CONDITIONS IN PITTSBURGH JUNE 12 TO SEPTEMBER 15, 1936



two age groups for lower temperatures, where a considerable feeling of coolness would pertain. Such tests could not be made in the short period available.

A considerable number of the men available for test appeared and entered the room without wearing coats, but otherwise dressed similarly to the trained subjects. These men show a measurably higher temperature requirement for comfort over that indicated for men wearing coats. The apparent difference is about 2, or possibly as high as 3 deg. The data collected in Toronto and plotted in Fig. 2 show about the same difference.

It was more difficult to get women of various walks of life and age distribution to enter the test rooms. The twelve women below 40 years of age

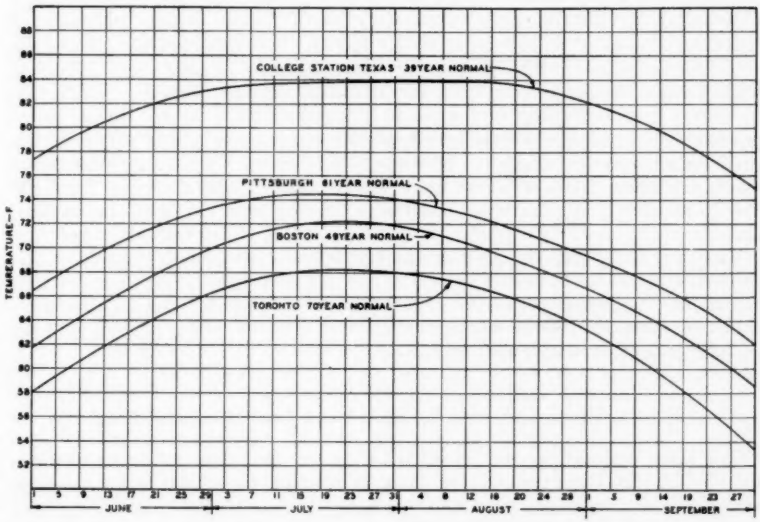


FIG. 16. NORMAL SUMMER TEMPERATURES REPORTED BY WEATHER BUREAU FOR TEXAS, PITTSBURGH, BOSTON AND TORONTO

entering the test rooms were employees of the Pittsburgh Station of the Bureau of Mines. They were largely members of the clerical staff. Their average indication of comfort was not materially different from that for men of the same age. Late in the season, or on the 17th and 18th of September, 40 members of the Bureau of Mines Women's Club, largely made up of wives, other relatives, and friends of the employees of the Bureau, arranged to enter the test rooms and give their opinion of the conditions. These arrangements had to be planned several days in advance and unfortunately the days on which the tests were made turned out to be rather cool, about 67 deg effective temperature. These women were dressed somewhat warmer than is customary for normal summer weather, and uniformly registered a degree of warmth of "4," or ideally comfortable, at 70 deg effective temperature, and slightly too

warm for comfort at 73 deg. Before entering the test room, these women spent about 15 min in a room at about 78 deg effective temperature, which accounts for the high degree of warmth registered before entering.

EFFECT OF AIR MOTION

A few tests were made with from three to five subjects to determine the desired temperature for comfort with higher than normal air velocities. These tests were made late in the season, and sufficient data for drawing any definite conclusion were not obtained. However, these few data serve to indicate that the effective temperature scale for moving air first published [3] by the Laboratory in 1924, for persons stripped to the waist, and later [4] corrected for persons normally clothed, applies reasonably well. These earlier studies were based upon tests at air velocities of 25, 150 and 300 fpm, and the effects for velocities between 25 and 150 fpm were determined by interpolation. The few studies made in this investigation for velocities ranging from 75 to 120 fpm indicate a greater cooling of from 1 to 2 deg than given by the earlier interpolated results, while for tests in velocities ranging from 120 to 300 fpm, the earlier published data appear to be correct. However, it should be emphasized that the number of observations made on the effect of air velocities were too few in number to be considered conclusive. Based upon the above indications, an air velocity of 75 fpm results in comfort with 82 F dry-bulb at 50 per cent relative humidity, and with 78 F dry-bulb at 80 per cent relative humidity. The subjects endured air velocities up to 150 fpm for periods up to one hour without complaint. Velocities above 150 fpm, even for periods of 15 or 20 min, were found objectionable, while velocities from 100 to 150 fpm for periods of over an hour became irritating and unpleasant.

SHOCK UPON ENTERING OR LEAVING A COOLED ATMOSPHERE

The plans outlined for the investigation by the Technical Advisory Committee on Comfort Requirements for Summer Cooling included a more intensive study of the cold shock upon entering a conditioned space, and the reaction upon re-entering the hot outside. Time was not available for any specific tests on this phase of the subject. However, occasional observations while making the studies reported above, and analyses of a few tests available at different effective temperatures, indicate quite clearly that the severity and length of the cold shock increases rapidly as the effective temperature of the cooled space is lowered, pointing to the desirability of maintaining as high an effective temperature indoors as will give comfort, or an effective temperature in the upper region of the comfort zone.

While few observations were made on the effect of the reaction upon re-entering the hot outside from the cooled space, it was apparent that the reaction decreased with a higher effective temperature maintained indoors. The few observations made, together with the general reactions and opinions of the

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[3] Cooling Effect on Human Beings Produced by Various Air Velocities, by F. C. Houghten and C. P. Yagloglou. (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 193).

[4] Effective Temperature with Clothing, by C. P. Yaglou and W. Edw. Miller. (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 89).

subjects taking part in the tests, indicate no very severe reaction, and it is believed that the objection frequently directed to air conditioning because of this reaction is largely psychological. Apparently, from the few observations made, recovery to complete normal reactions after re-entering the hot outside (if such reactions can be considered normal), was reached as soon as perspiration was re-established. Upon re-entering the hot outside, the reverse in physiological reactions takes place from those upon entering the cooled space. Before leaving the cooled space, a person's heat dissipation is in equilibrium with the cooled condition, and the rate of heat production and adequate heat dissipation results without sensible perspiration. As soon as this person re-enters the hot atmosphere, equilibrium is destroyed and the body ceases to dissipate heat to the atmosphere at the normal rate, because sensible heat dissipation has been decreased with no compensation by increased evaporation. Equilibrium, and hence, a normal feeling of warmth, does not result until normal perspiration for the hot condition has been re-established. A few observations of this reaction indicate that from 5 to 15 min is required to re-establish perspiration. This time is apparently shortened by exercise.

The subjects never complained of discomfort during the hours following a cool test other than for the short period mentioned above, excepting that their greater comfort during the test period emphasized the discomfort resulting from the hot outside atmosphere.

REVIEW OF THE RESULTS OF THE STUDY

The study of the cooling requirements for comfort in summer air conditioning in Toronto, Texas and Pittsburgh confirm the findings of a year ago that comfort may be had over a small effective temperature range, which is independent of daily or even weekly variations in outside weather. The results show clearly that with the weather conditions of last July in Toronto, an effective temperature of from 2 to 3 deg lower was required for comfort than was the case in Pittsburgh. Similar studies made in Texas prove rather conclusively that for their hotter, more continuous and longer summer heat, the same effective temperature is required as was found for Pittsburgh. In Toronto, 70 per cent of the subjects were found to be comfortable over a range of effective temperatures from 68.5 to 71 deg; in Pittsburgh from 69.5 to 74 deg; and in Texas from 71 to 74 deg.

In order to minimize the cold shock upon entering a conditioned space, as well as to economize in the cost of cooling, an optimum effective temperature for air conditioning purposes is recommended near the upper limits of the comfort zone, or 71 deg effective temperature for Toronto, and 73 deg for localities having summer conditions as hot or hotter than Pittsburgh.

An intensive study of the relation of comfort to relative humidity with constant effective temperature indicates little or no measurable variation in the time required for perspiration to disappear and for comfort to be established in a relative humidity range from 20 to 90 per cent. However, a recommendation for the use of relative humidities of 80 per cent or higher must be given with reservation, due to the difficulty experienced in properly supplying these higher humidities in the summer with the air conditioning system available.

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Sixty-one men, ranging from 20 to 65 years in age, were found to be comfortable over approximately the same temperature range as the trained, young subjects. A slightly higher temperature was indicated as desirable by men over forty. Few data were obtained for women of different age groups under satisfactory weather and environmental conditions, but these few observations indicate no material difference in the degree of cooling desired over that found for men.

Clothing was found to have a material effect on the desired temperature for comfort. The wearing of a lightweight coat apparently decreases the temperature required for comfort by from 2 to 3 deg. Hence, the removing of such a coat raises the maximum temperature for comfort by 2 or 3 deg.

Acknowledgment is due to the Canadian Westinghouse Co., Ltd., the Minneapolis-Honeywell Regulator Co., Ltd., the C. A. Dunham Co., Ltd., and the Canadian General Electric Co., Ltd., for their cooperation in the loan of the equipment used in the tests made at the Ontario Research Foundation, Toronto.

DISCUSSION

W. E. STARK (WRITTEN): It is probably true that there is no human sensation more intangible and difficult to evaluate than that of comfort, because it is so intertwined with the mental and physical state of the subject, kind of clothing worn, and possibly with preconceived notions. The sensation of warmth, or the feeling of bodily coolness or warmness, is more tangible and has a less pronounced relation to the mental and physical state of the subject, and is therefore easier to evaluate. The fact that warmth is more susceptible to evaluation than is comfort is probably the reason for the seeming tendency to consider the two as being one and the same thing and to regard the sensation of warmth as being synonymous with the sensation of comfort.

In THE GUIDE we find these words: "Effective temperature is an index of warmth or cold. It is not in itself an index of comfort, as it is often assumed to be, nor are the effective temperature lines necessarily lines of equal comfort. . . . Moist air at a comparatively low temperature and dry air at a higher temperature may each feel as warm as air of an intermediate temperature and humidity, but the *comfort* experienced in the three air conditions would be different, although the effective temperature is the same.

Under extreme humidity conditions there seems to be a difference between sensations of absolute comfort and of the proper degree of warmth. In other words, human beings are not necessarily comfortable when the air is neither *too warm* nor *too cold*. . . ."

I believe those words and accept them as sound air conditioning doctrine. However, I am now asked to believe that with constant effective temperature comfort can be established in a relative humidity range from 20 to 90 per cent. In other words a group of subjects have registered their sensations of warmth, incidentally verifying a portion of the effective temperature chart; but warmth and comfort have become confused and practically the whole length of a constant effective temperature line has become a constant comfort line. I cannot believe in this.

Anyone unfamiliar with air conditioning practice and the results thereof, and reading this paper in search of information, would finish it with the conclusion that

in Pittsburgh and College Station, Tex., and presumably at all intermediate points, an effective temperature from 71 to 74 deg with relative humidities ranging all the way from 20 per cent to 90 per cent will make at least 70 per cent of the occupants of a conditioned space comfortable and that if an occupant steps out of one extreme humidity condition into the other extreme humidity condition, he would not know the difference. Now let us set up some of the properties of air under these extreme conditions in tabular form and compare them. We can make up the accompanying table, in which the last column is intended to represent what might be looked upon as an average outdoor condition:

EFFECTIVE TEMPERATURE	71 deg		74 deg		81.4 deg
Relative humidity	20%	90%	20%	90%	50%
Dry-bulb	80.5 deg	71.9 deg	85.3 deg	75.0 deg	90.0 deg
Wet-bulb	56.6 deg	69.7 deg	59.8 deg	72.8 deg	75.0 deg
Total heat	24.1 Btu	33.7 Btu	26.2 Btu	36.4 Btu	38.46 Btu
Grains per pound	31.0	106.0	36.0	118.0	106.0
Dew-point	36.0 deg	68.7 deg	39.7 deg	71.9 deg	68.9 deg
Body heat loss	400 Btu	400 Btu	400 Btu	400 Btu	400 Btu
Per cent sensible loss	54%	72%	41%	66%	27%
Per cent latent loss	46%	28%	59%	34%	73%

If it is true that any condition within the range of the first four columns of this table represents comfort, why are there any poor air conditioning jobs?

Certainly the human mechanism is not so flexible that it can have a latent heat loss anywhere between 28 per cent and 59 per cent of its total heat loss and still not have the difference reflected in comfort. There may well be a wider range in relative humidity than we have conventionally thought, over which the sensation of comfort is practically invariable, but it does not seem sound to believe that wet-bulb temperatures can fluctuate between 56.6 and 72.8 deg without the body noticing the difference.

The statement is made in the paper that there is no relation between the cold shock when entering a conditioned space and the relative humidity maintained at a given effective temperature. It would seem to be rather sound reasoning to believe that the less the relation between the two components of the heat loss from the body (sensible and latent) is disturbed upon entering a conditioned space, the less severe the shock should be. Since the existing data in the THE GUIDE indicate that it is dry-bulb temperature that determines the relation between these two components of heat loss, the conclusion of this line of reasoning is that shock would be minimized by placing more emphasis on the moisture or humidity reduction than on the temperature reduction when striving to reach a given effective temperature. The same thing ought to be true on leaving the conditioned space.

In other words, the less adjustment that the skin is obliged to make in the rate at which it loses heat by evaporation of moisture, the less the shock should be. The magnitude of the adjustment that the skin must make may be reflected in health as well as in comfort.

The people who are best qualified to tell us what conditions are comfortable are the paying customers; those who pay to eat in an air conditioned restaurant, to see a show in an air conditioned theater, or to shop in an air conditioned store. The reactions of those people, most of them totally ignorant of such things as effective temperature and relative humidity, are more sound indicators of what constitutes comfort than are the reactions of a group of college students, all of them more or less familiar with the subject and acutely aware of what is going on. Such people, quite ignorant of the technicalities of air conditioning and speaking spontaneously, not after reflecting on whether their feelings are described by *3*, *4*, or *5*, remark

that air conditioned places are too cold and damp, or hot and muggy. I think such spontaneous expressions are much more reliable indices, and I have an idea that the subjects who are said in the paper to have indicated their displeasure with high humidities, although they admitted it had nothing to do with their feeling of warmth, were really giving vent unconsciously to their spontaneous, but unscientific and therefore more human, reactions.

OKLAHOMA CHAPTER[5] (WRITTEN): With regard to the discussion on the Laboratory paper, a number of the members expressed themselves as surprised that the research had not shown a greater effect on the variation in the humidity. Locally, it has been found that 73 deg ET seems to be most comfortable for the majority of the occupants when the temperature is maintained at 82 deg with a 30 per cent relative humidity, but that when the relative humidity is permitted to rise much above that figure, complaints of being too warm greatly increase. It was noted also that practically all the laboratory tests have been made on young, healthy subjects, whereas in commercial work air conditioning has to deal with all ages and physical conditions, so that the work should be continued to find the effect on other types of subjects.

The matter of shock is also of great importance as in the sale of air conditioning equipment, one of the first objections usually heard is that people experience such an unpleasant reaction in going out of a conditioned building into the hot street. Some of the members believe that this shock is largely psychological, and that as people become more accustomed to air conditioning, we will hear less about it; while others testified from their own personal experience that this shock is really physical and further study is needed to determine just what takes place.

It would seem that some study could be made to find out more definitely just what happens when a person does go out of a conditioned room into the heat. For instance, is there a rapid rise in the body temperature? Does the heart action speed up, and is this effect alleviated as soon as perspiration starts? In other words, is the lag in evaporative cooling one of the reasons for this shock? One member stated that the chief engineer of the Mayo Brothers Clinic had told the Kansas City Chapter some time ago that no actual physiological shock had been found, whereas another member stated he had knowledge of a store in Kansas City which had found it necessary to have a nurse on duty to care for those who suffered from shock. It was the opinion of others that fainting, which has sometimes been attributed to this shock, is probably due to the excessive heat and the same subject going out into the hot street from a building that was not air conditioned would experience the same reaction.

It is hoped that much additional work will be done in studying these various factors as they cover some of the most important phases of the air conditioning field.

G. B. HELMRICH (WRITTEN): As many of you will recall, The Detroit Edison Co. has sponsored experiments in residential cooling for the past five summers and most of these experiments have been carried out in a residence in Birmingham. Several papers have been presented at A.S.H.V.E. meetings summarizing the results of these cooling experiments and a correlation of our experience with that recounted in this paper should be of some interest.

We have cooled our experimental residence with a 2-ton refrigerating machine during the past three summers and we have found that the most comfortable effective indoor temperature downstairs lies between 72 and 73 deg, corresponding to dry-bulb temperatures of 77 to 79 deg, and relative humidities of 60 to 50 per cent. This

[5] Presented by E. W. Gray, Secy., Oklahoma Chapter.

temperature lies between the temperatures given in this paper for Pittsburgh and Toronto, and, therefore, shows a nice correlation between our experience and theirs. We have found out fairly definitely that an effective temperature of 74 deg is a little too high for comfort, except under extreme conditions of 100 deg, or higher, outdoor temperatures.

I think there is some tendency in this paper to over emphasize the matter of cold shock, or the feeling of discomfort upon entering a cooled atmosphere. I well remember having occasion to drive out to Birmingham on a hot Sunday afternoon, when the outdoor temperature was over 100 deg, to visit our Birmingham residence to see how well the cooling equipment was functioning. The living room temperature was 80 F and the relative humidity 50 per cent, the effective temperature 73.8 deg. I experienced no shock upon entering the house under these conditions, but, on the contrary, found it delightfully cool and comfortable and stayed considerably longer than I had intended to. The occupants of this home have, however, upon a few occasions, experienced a slight feeling of clamminess for a period of 5 or 10 min when they entered the house after a long journey from downtown during which they had perspired rather freely.

I question the practical value of the experiments in which the effective temperature was kept constant at 73 deg, while the relative humidity was increased to as high as 90 per cent. 73 deg ET and 90 per cent relative humidity call for about 74 deg DB temperature, and I cannot believe that this condition can be as comfortable as that existing when, for the same effective temperature, the dry-bulb is around 77-78 deg and relative humidity 50-60 per cent. I believe a condition of 90 per cent relative humidity and 74 deg DB is bound to be somewhat cold and disagreeable, although I realize that no claims were made that above 70 per cent relative humidity the same effective temperature necessarily produces equally comfortable conditions. I do believe, however, that reference to these extreme conditions is misleading to some readers. I would like to see the A.S.H.V.E. Laboratory at Pittsburgh do more experimental work in the way of checking effective temperatures in the comfort zone above 70 per cent and below 30 per cent relative humidity. I am not at all ready to accept the present experimental data for either top or bottom of the comfort chart.

With regard to the tests conducted with air velocities as high as 150 fpm, I wonder how many of us realize the rate of air change required, and size of fan involved, to produce such velocities. We had occasion to test the cooling power of the furnace fan this past summer by measuring air velocities and cooling power with a Kata thermometer. In the living room of our experimental home we have air leaving two base board registers at an average velocity of 300 fpm and the Kata thermometer showed an air movement in the room, at the 30-60-in. level, not to exceed 40 fpm. As the rate of air change for this room is about 7½ per hour, it is obvious, that to produce an air motion in the room equivalent to 150 fpm, the air change rate would have to be increased to about 21 per hour. This would require the installation of about three times as much fan capacity as is now usually supplied by the manufacturers. It is the present practice of mechanical warm-air furnace manufacturers to supply fan capacity for four to six air changes per hour. I therefore question, at least so far as residences are concerned, the practical value of testing for the effect of air motion beyond velocities of 50 fpm.

PROF. F. J. LINSENMEYER (WRITTEN): The paper has real significance in the study of comfort air conditioning inasmuch as three independent experimenters, each with the accumulated experience of three such tests previously carried on, were able to check last year's conclusions and agree among themselves so well.

Two simple, but important conclusions stand out clearly; *first*, that there is such a great difference in July averages between Toronto and College Station—67 and

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83.5 F, which seems to require 24-hour air conditioning in Texas and cyclic conditioning in Toronto; *second*, in spite of the importance given heretofore, the *almost* constant effective temperature requirements in all sections of the country.

The paper states the conditions under which the tests were made; but one must not lose sight of the adjustments that should be made for the practical case of hot walls. With the solar-radiation on the walls it may be expected that about the same adjustment downward would be needed as is required upward under winter conditions; namely 2 deg ET. However the discusser quite fully realizes the need for establishing some agreement under cool-wall conditions during this stage of the experimenting.

It is to be expected that the subjects tested were familiar with the conditions of the test and nomenclature, and were psychologically disposed to a fair interpretation. Right here there is a difference between these subjects and the customers or the public. The public is not sufficiently conscious of the variables and too much swayed by hearsay to be a satisfactory judge of comfort. Accordingly, conclusions as to the possible disagreement of the present test results and practice should be reserved for further education of the public.

There is one chance for question in the present group of tests, namely, the comfort condition is referred always to effective temperature without going back over the proper interpretation of "effective." Effective temperatures are based on a set of tests covering equal warmth which is in a sense what the present test is doing also. It would seem desirable to check independently against the wet- and dry-bulb, and air motion with respect to comfort.

By examination of Table 1, it is interesting to note that the subject reaches equilibrium with respect to perspiration within an average of 20 min after entering the conditioned room, and that the range of comfort is only about one point from comfortably cool to cool.

Regarding the difficulty of supplying relative humidities in the higher range, there is not such great difficulty in the use of well water for cooling and it is encouraging for this form of cooling to note that it is the most economical—that is, less cooling is needed for sensible heat removal than for latent heat.

MINNESOTA CHAPTER[6] (WRITTEN): Before a definite conclusion is drawn as to the following statement, "An intensive study of the relation of comfort to relative humidity with constant effective temperature indicates little or no measurable variation in the time required for perspiration to disappear and for comfort to be established in a relative humidity range from 20 to 90 per cent," care must be exercised that this statement is not misinterpreted by the non-technical man. It is very easy to overlook the fact that effective temperature is not a temperature but an index as to the degree of warmth and comfort felt by the human body. For example taking the upper and lower limits of 73 deg ET and 90 per cent and 20 per cent relative humidity, the corresponding dry-bulb temperatures are 74 deg and 84 deg respectively or a difference in dry-bulb temperature of 10 deg.

This paper stresses the feeling of warmth felt by the human body, not taking into consideration comfort and health which are just as important. There is a distinct difference between warmth and comfort as experienced by the human body, the first being a physical reaction and the second physiological. When the feeling of warmth has been experienced by the human body, a balance has been set up between the rate of heat generation in the human body and that dissipated to the surrounding air. The second or physiological reaction is the effect upon the health of the body. As

[6] Presented by N. D. Adams, Pres., Minnesota Chapter.

is stated in this paper, "importance was placed primarily on its effects on the feeling of warmth," referring to the effect of relative humidity.

Prof. C. P. Yaglou of the Harvard School of Public Health, in his paper on Physiologic Changes During Exposure to Ionized Air, found that ionized air appeared to exert a normalizing influence upon the organism by accelerating the physiologic processes in instances in which these processes were below the normal range of the majority of subjects and, reversely, by decreasing the physiologic activity in cases in which the functions were above the normal range.

Prof. C.-E. A. Winslow, director of John B. Pierce Laboratory of Hygiene, has found that varying ion content of outdoor air has a distinct effect upon the comfort of the human body. He also states that the following relationships are apparent for both positive and negative total ions: "Total ions increase with increase in relative humidity and barometric reading".

In an article on Air Conditioning for Health, Professor Winslow makes the statement: "Evidence continues to accumulate as to the important influences of air temperature upon human health, comfort and efficiency. From the standpoint of humidity we know that very dry or very moist air is undesirable although the A.S.H.V.E. studies at Pittsburgh suggested that, over a rather wide median range, variations in relative humidity are not of major importance".

From the foregoing discussion, it appears that relative humidity does have an effect upon the comfort and health of the human body and that further study should be made of other factors on the basis of comfort and health before any definite conclusions are drawn.

The outside design temperature for cooling in Pittsburgh is 91 deg DB and 73 deg WB. Using 73 deg ET inside with varying humidities, the dry-bulb temperatures will vary 7 to 17 deg depending upon the relative humidity maintained. According to Dr. Lloyd Arnold, professor of Public Health and Hygiene, University of Illinois Medical School, the medical opinion both in this country and abroad is that the difference between inside and outside temperature on a hot day should not be more than 10 F or possibly less, and the inside humidity should probably be between 40 and 50 per cent with air velocity an important factor. This is to eliminate the harmful effects of sudden temperature changes thereby preventing intestinal upsets and head colds.

It would have been interesting to know just how many subjects were tested. The percentages do not indicate that. It makes a great deal of difference whether 75 per cent is represented by eight, 50 or 750. Also are the ages 19-25 years used for this study representative? The paper mentions 61 men between the ages of 20 and 65 and does mention 12 women under 40 and 40 women without mentioning their ages. It also states that it was found that there was no appreciable difference in the data obtained from this miscellaneous group of people than for the trained subjects, but that "men" over 40 need a little more heat. What about the women over 40? No doubt theaters during the hottest part of the day are visited mostly by women, and very likely women over 40. It is suggested that large theaters be enlisted to help with this investigation. The whole scheme would be explained to the patrons on the screen and the reactions might be indicated by cards handed to patrons. In this way various conditions could be artificially created and a vastly larger number of tests could be made.

The paper is most interesting but does not prove conclusively that the effective temperature is all that is necessary to take into account in the designing of summer air conditioning.

True, a feeling of comfort must prevail in the conditioned space and these data

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show an effective temperature of approximately 73 deg as being proper, depending upon the geographical location.

Much more conclusive proof must be shown before we can accept the effective temperature line as the only guide to good design. It would be very unwise to accept this conclusion and recommend the expenditure of large amounts of money for equipment which later might have to be discarded because of its ineffectiveness.

Medical men have not been heard from as much as they will be in the future. It is more important to build for the physiological well being than it is for the psychological. Medical opinion has expressed itself in the past in favor of high dry-bulb and low wet-bulb temperatures. These opinions although not conclusive cannot be disregarded.

Dr. Wm. H. Welker of the University of Illinois suggests a dry-bulb temperature of 80 deg with 30 per cent relative humidity as the most satisfactory for hay fever patients.

L. W. Wallace, director of Research for the *American Railway Association,* says that in his opinion summer air conditioning must be designed to keep away from the cold clammy condition experienced in so many installations and definitely control both wet- and dry-bulb temperatures.

The data in this paper are far too incomplete to be accepted, until further study is made on the physiological effect.

PHILADELPHIA CHAPTER[7] (WRITTEN): This paper presents the findings in a type of research that has been greatly needed in the field of air conditioning. The conclusions are a real contribution to the recorded knowledge on the subject. However, there are certain phases of the investigation that we believe should receive further study.

The recommendations of a minimum effective temperature of 69.5 deg in summer for climatic conditions such as exist in Philadelphia seems to be more satisfactory than the 68 deg as recommended in the present issue of THE GUIDE.

The statement appears in the third paragraph on page 153 "That no variation in the desired indoor cooling temperature accompanies a daily or even weekly change in outside weather, excepting possibly on days when the outside is cooler than the inside". If this statement is to be interpreted that when a system is designed to hold 73 deg ET when outside conditions are 95 deg DB and 78 deg WB that they shall continue to maintain 73 deg ET inside until outside conditions fall below this value, then additional study should be given before departing from the scale of graduated differences appearing on page 73 of THE GUIDE, 1936, as the present practice appears to be more satisfactory.

The research paper indicates that approximately 90 per cent of the people are comfortable with an effective temperature of 71 under maximum outside conditions. If this is correct we believe that in practice comfort cooling systems should be operated to maintain an effective temperature of 71 the greater portion of the time. However actual practice has indicated that 80 deg DB and 50 per cent relative humidity is comfortable in commercial establishments under outside extremes of 95 deg DB and 78 deg WB. This design condition has gained public acceptance. We believe that additional studies should be made before recommending that systems be designed to maintain an effective temperature of 71 on a 95-78 day. Such change

[7] Discussion prepared by a committee of the Philadelphia Chapter consisting of the following personnel: H. H. Mather, *Chairman,* A. E. Dambly, P. L. Davidson, H. P. Gant, C. S. Leopold, L. Mack, H. G. McCullough, L. E. Moody, A. J. Nesbitt, G. E. Tuckerman, and presented by H. H. Erickson, Secy.

in design conditions would require considerable educational work to achieve public acceptance.

We believe that further research should be conducted on humidities above 60 per cent before such conditions are recommended for actual practice. Past experience would indicate that humidities in excess of 60 per cent do not result in a sensation of comfort. Investigations should also be conducted to determine the effect of high humidities on the merchandise and various types of fixtures and appointments usually encountered in commercial establishments.

C. S. LEOPOLD (WRITTEN): Under note—"depends on geography," Fig. 6 shows the result that might be reasonably expected: namely, that people conditioned to normally cool atmosphere can be comfortable to the cold side and not adaptable to the hot (Toronto), and that people conditioned to the hot side (Texas) can stand the extremes in hot but not the extremes in cold; but that people conditioned to the erratic climates of Boston and Pittsburgh can stand either extreme almost as well as the people who were specially conditioned to one of these extremes. This is not surprising in view of Dr. Mills' work on rabbits.

I would suggest that you note that these tests were mostly made for sensation after one-half hour, under which conditions these data should be, and to the best of our experience are, accurate. We caution, and we think the article should caution, against exterpolating these data.

We believe Mr. Houghten passes too lightly over the shock at exit. General experience is that for people in normal health the shock on entrance is relatively pleasing but that the shock of exit is often distressing, particularly as some people experience momentary breathing difficulties which are usually accompanied by fear, which tends to produce the psychological result Mr. Houghten speaks of, but I do not think a fear so induced can be dismissed as a purely psychological effect.

We have weather data that would tend to indicate, though by no means conclusively, that in normal individuals the change from warm to cool is stimulating and probably healthful, and that the converse is true on the change from cool to warm. We believe that this exit shock warrants a great deal more study.

CINCINNATI CHAPTER[*] (WRITTEN): 1. Relation Between the Desired Effective Temperature for Summer Comfort Air Conditioning and the Weather Conditions of Different Geographical Locations: It was the consensus of opinion of those present that the conclusions of the paper, recommending varying effective temperatures for various locations, were correct. Instances were cited where it had been necessary to carry higher effective temperatures in states in the south and southwest than are normally carried in northern territory and that complaints of overcooling were eliminated by carrying higher effective temperatures. There was also agreement on the recommendation of 73 to 74 deg ET for local territory. One member stated that he was normally designing for 74 deg ET with satisfactory results in every instance.

2. Variations in Relative Humidity at a Given Effective Temperature: This section of the paper created the greatest interest and brought forth the major discussion of the evening.

There was general agreement that higher humidities than 50 per cent could be reasonably used for comfort cooling, although it was felt that humidities of 80 to 90 per cent would be impractical. It was mentioned that extremely high relative humidities would possibly have a detrimental effect upon decorations, woodwork, and furnishings. One member stated that he had observed increasing difficulties from

[*] Presented by O. W. Motz, Secy., Cincinnati Chapter.

odors as the relative humidity was increased above 55 or 60 per cent. It was suggested that this could possibly be due to lack of cleanliness in connection with the air washers or surface coolers; and it was then agreed that these factors might also have contributed to the odors in the specific instance mentioned.

An instance was cited where considerable economies in refrigerating and conditioning equipment were made by increasing the design relative humidity from 50 to 55 per cent, thus raising the dew-point and water temperature to a more practical design point.

It was mentioned that maintaining higher relative humidities would require higher air motion and the circulation of larger air quantities. It was felt that more specific information should be included in the paper on the air motion prevailing during the tests, since it is a major contributing factor in determining the effective temperature. Thus, the same effective temperature could possibly be obtained with high air motion, and evaporative cooling, rather than by mechanical cooling and dehumidification.

There were two schools of thought on daily variations of inside conditions in accordance with the daily variation in outdoor temperature; one being that the inside conditions should be constant, as recommended in the paper, and the other being the inside conditions should be varied. One specific example was cited, in an office building, where complaints were registered when the conditions were increased with the weather; the complaints were eliminated by maintaining fixed, lower conditions of approximately 78 to 80 deg DB, and 50 per cent relative humidity. The Cincinnati Board of Health is considering legislation requiring inside conditions to be limited in accordance with outside conditions. It was felt that such action was not warranted in view of the facts presented.

The paper was highly praised by all present as being an outstanding and very timely contribution of the Research Laboratory, and as demonstrating the real and practical value of work of the Society. We hope that these studies will be continued.

DR. D. L. MACLEAN[9] (WRITTEN): This discussion has been prepared by Dr. Partridge and myself in regard to the summer comfort values. The value of 70 deg ET for the summer comfort line obtained by Mr. Tasker is in agreement with a value of 70.5 deg ET obtained by MacLean and Partridge in Toronto in 1933. In the MacLean-Partridge survey 70 per cent of the subjects were found to be comfortable in the summer over a range of effective temperatures of 67.5 to 74 deg. This range is wider than that reported by Mr. Tasker. The upper value for the MacLean-Partridge study was the same as now reported for Pittsburgh and Texas (74 deg), while the lower value of the range was 67.5 deg for Toronto, 69.5 deg for Pittsburgh and 71 deg for Texas.

The MacLean-Partridge survey was carried out with only six adults, a number which they felt to be small for any biological research. The results for adults were reported, however, since they were in agreement with their results when 25 children were used and 25 to 100 observations for each effective temperature value obtained. Their results show that the comfort line and comfort zone for children in the summer is the same as for adults, although it might be suspected that children whose metabolic rate differs from adults would require different values for comfort. Their results show, moreover, the effect of acclimatization since the adults who were wearing similar clothing in both seasons had a lower value for maximal comfort in the winter, namely 66.5 deg ET.

In addition to the acclimatization factor clothing affects the results. The effect of clothing was shown by their observation that the range of comfort in the winter for the boys, who were most heavily clothed, was wider than that for the girls or

[9] Dept. of Physiological Hygiene, University of Toronto.

the adults. In the summer the comfort zone and comfort line were the same for boys, girls and adults, hence the above mentioned difference was not due to age or sex factors. It is significant that the difference in clothing does not change the value for maximal comfort, for the comfort line was the same for boys, girls and adults, nor does it change greatly the upper limits of the comfort zone. Its chief effect seems to be on the lower value for the comfort zone range. Those who were heavily clothed were better able to withstand the cold temperatures. Since the above study was carried out with subjects under 35 years of age, a further investigation of the effect of age and occupation on the comfort zone is contemplated.

DR. C. A. MILLS: I am doubtful as to the advisability of considering Pittsburgh and any part of Texas on the same level just on the basis of these observations, since they were made during the latter half of the summer (July), when Pittsburgh had been having temperatures considerably above those of Texas. Here arises the question of adaptation in internal heat production that I have many times stressed before you in recent years. Two weeks, or longer, of severe heat will cause the body to reduce its own heat production. Hence Pittsburgh in this past year (1936), a little less in 1935, but particularly so this year, has had a population that was compelled to adapt to a temperature level fully equivalent to that of Texas. This did not hold for Toronto with the brief heat it had for only two or three days. With two or three days of outside heat people will desire for comfort the same indoor temperature they had previously desired, but if they have severe heat for two or three *weeks,* their comfort level rises to approximately that of any other people subjected to such prolonged heat.

The mean or normal temperatures which Mr. Houghten gave in the curve for the three cities, showing such a wide spread between Texas and Pittsburgh, should be considered more applicable to the difference in the requirements for comfort for the people of the two regions, rather than comparisons made during severe heat at Pittsburgh, heat often more severe than in Texas. The figures taken in Boston for 1927, when the summer temperatures of the whole continent were considerably below the temperatures of the last couple of years, brought Boston nearer the present Toronto level comfort than the mean curve seems to indicate.

What I wish particularly to stress is that, in attempting to establish standards for installations in different regions, the data to be used as representative of a given region should be collected under normal conditions for that region. Severe heat in Pittsburgh, if sufficiently prolonged, will surely bring about a rise in comfort zone for Pittsburgh inhabitants up to the Texas level. Or Toronto at the end of a hot summer may have a comfort level above that of Boston after a cool summer. Let us, then, base standards for the range to be supplied by installations on findings obtained from test subjects under normal condition for the regions in question, and not on the occasional wide departures from that normal.

W. L. FLEISHER: May I say a word on this interesting subject? I want to bring to the attention of the members the fact that the Laboratory is not recommending that people should use 90 per cent relative humidity any more than 30 per cent relative humidity. The general discussion and criticism of the work that has been done by the Laboratory has been of the usual wish fulfillment variety rather than actual facts.

Over a period of two, three or four years during which I have been associated with Mr. Houghten on this work, there has been definitely no attempt to dictate to the members or to the Society, the use of any particular point on an effective temperature line. The whole investigation really started with an idea of differentiating between the so-called effective temperature line and a constant dew-point line, because under certain conditions where chemicals were not utilized, it took considerably more

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refrigeration to maintain the constant dew-point condition than certain other conditions on this particular line, and, of course, originally, this particular line originated from the fact that to continue on this effective temperature line down to the lower relative humidities, required practically twice as much refrigeration as at the 57 deg dew-point. With chemicals, dehumidifiers or absorbers, this condition could be reached quite as readily as other conditions along this particular line with regulation.

It is astonishing that over two or three years of repeated investigations it has been definitely indicated that at relative humidities decidedly above those which had been considered as comfortable, people have felt comfortable, notwithstanding any prejudices that exist for low humidity or dry conditions, and, in many cases, the reactions of people brought in contact with high humidities evidenced less shocks than lower humidity conditions on this particular line. This was true particularly when perspiring.

Now, a schedule of figures such as have been shown, although they look fairly important, are to my mind of no particular significance. Nobody practices using that point, and nobody is using that particular point, and the dissipation of heat from the body in accordance with these proportions must certainly have originated from statistical tabulations and not from actualities. It is astonishing that visible perspiration which, in a way, is a measure of comfort, in that until visible perspiration disappears, your subjects don't express a feeling of comfort, does disappear almost as rapidly in high humidities as in the lower ranges.

I am sorry that Mr. Houghten did not bring that out. It is one of the most interesting parts of the work done this summer and observed by me in my own laboratory, and is due to the fact that the skin temperature underneath the visible moisture has a tendency to evaporate that moisture irrespective of the conditions that surround the body, whenever the air motion, however slight, may carry off the moisture as a vapor after being evaporated by the temperature of the skin.

As I say, it is extremely important that data which have been carefully taken, should be given more importance than the general opinion or feeling of the Society, which is based on ideas not determined empirically. As Dr. Sheard said, you can't take people's ideas of what is comfortable or not comfortable. I have run hundreds of tests in which I changed the conditions to exactly opposite what people thought were comfortable and they didn't know the difference.

As to actual experience with morons, I think that is a very good method of finding out things. I had the opportunity to investigate the reaction of morons in 20- to 25-cent restaurants over a period of three years, who were subjected to conditions of over 75 per cent relative humidity on these effective temperature lines, and some 17,000,000 people occupied these cheap restaurants for short periods, an average of 25 to 30 min, and the reaction was sufficiently good so that, in a large chain of restaurants, the restaurant owners are continuing that practice. I therefore say that experience which is gained by laboratories, established well-conducted laboratories, must be given more credence than the generally loosely expressed opinion of the public.

DR. E. V. HILL: Mr. President, I do not wish to take up very much time as it is getting late but I cannot refrain from making one observation that has not been touched upon here today. You heard Dr. Sheard tell about the work of a famous physician, Dr. S. Weir Mitchell, who attempted to plot the barometric pressure against the sensation of pain, and he told us what a hopeless task it was. He also indicated, and I have the greatest respect for Dr. Sheard's opinion based as it is on many years of very careful research, that the sensations you elicit by questioning a subject are practically worthless.

Now, if you go back with me to 1920, when the effective temperature lines were plotted, you all know they were plotted from data obtained by that method of procedure, by asking subjects how they felt, and you learned today, if you listened carefully to Dr. Sheard, that the temperature of the hands and feet, the clothing, the time before and after meals, and a hundred other things affect the feeling of comfort.

I have said many times, although I never get very far with it, and I don't expect to today, that the effective temperature zone as used by the Society must be entirely explored. I am satisfied, in my own mind, that the effective temperature lines are not straight lines on the Carrier Psychrometric Chart. This leads us to another point, as to what possible value has this indeterminate thing, the effective temperature. Even if it were true we could not use it for psychrometric data, like the wet-bulb, the dry-bulb, and dew-point, being in itself a variable with an indefinite meaning.

In this work that Mr. Houghten has been carrying on, he has plotted a thing that is at least questionable, effective temperature against subjective sensation, which we all know, if we have gone into it at all, is practically worthless. So, you are plotting an indeterminate against a pipe dream, and that explains the number of the points on the chart, which John Howatt has called the "fish roe" chart.

I think that the work ought to be reanalyzed; I think that all the results ought to be plotted against known psychrometric values, the dry-bulb and wet-bulb and dew-point, that we could understand. In saying this I hope you all realize that I am a life long friend of Mr. Houghten and expect to continue to be, and have the greatest regard for his honesty, his technical methods and his opinion but I also think that this effective temperature has led us into a maze of—what can we call it? I can't think of a name for it—but something that is meaningless and getting us nowhere and that sooner or later, and I hope it will be sooner than later, the Society will have to discard and forget it.

I might just remark, before sitting down, that at the first meeting of the Research Committee, held in Pittsburgh, when our first Director was selected some 18 or 20 years ago, the question was asked "what will be the first problem that the Research Committee should undertake" and I suggested "the comfort zone." They have been working on it ever since and getting further and further away from the true answer.

C. TASKER: I would like to point out that several years ago some work was done by the Department of Physiological Hygiene in Toronto by which a maximum comfort line of 70½ deg ET was established for both adults and children. This checked very well with the 70 deg we got this year. There is one thing that I wish Dr. Mills could tell us for it is worrying me quite a lot. There is only a 6 deg difference in those normal summer temperatures on the chart between Toronto and Pittsburgh, and yet the tests showed a difference of almost 3 deg ET for our requirement for maximum comfort; whilst there is a 10 deg difference between Pittsburgh and Texas in the normal summer temperatures on the chart with, apparently, no significant difference in the location of the required comfort line. This thing worried me very much when the results first came through, and I am afraid is going to continue to worry me until some one can explain it.

PROF. F. E. GIESECKE: The work at College Station, Texas, and at Toronto was carried on as nearly as possible like the work at Pittsburgh. The results were studied and co-ordinated by Mr. Houghten and his associates, and we have asked Mr. Houghten to close the discussion.

I would like to add that the usual difference of 10 deg between the average summer temperature in Texas and the average summer temperature at Pittsburgh did not exist when the tests were being made, because the tests in Texas were made

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too early in the summer. For this reason, we believe that additional tests should be made and we expect to do that in Texas throughout the coming 12-month period and to correlate the results with the corresponding out-door conditions so far as possible.

F. C. HOUGHTEN: The effective temperature idea, as Dr. Hill calls it, or the effective temperature chart, sometimes called the comfort chart, as established by your Laboratory may, conceivably, be proven to be in error in some minor respects. Although it has been reviewed critically in all parts of the world since its first presentation, no such critical study of it has so far been able to establish even minor errors. But even accepting that some minor errors may be found in the future, I am as certain, and I am sure that all who have associated with me in this work agree, that the principal facts established in that chart, or in other words the location and slope of the effective temperature or comfort lines, will never be proven to be very far wrong. Any conceivable adjustments in those lines based upon future research are freely predicted to be smaller in magnitude than the present state of the profession can detect or use in practice. Although we are thus confident in the soundness of the Society's comfort chart we must, and do welcome, all honest criticisms or checks which any one wants to direct toward it, and, of course, this same point of view must prevail as regards all other work of your Research Laboratory.

Some of the discussion of this paper has dwelt upon a presumed important difference between a person's feeling of warmth and his well-being or health. Without much thought, medical men frequently raise this question, but when asked what better measure or criterion they have to offer for a person's well-being or health they invariably admit that there is none, and until some other superior measure or index is developed we must accept a person's comfort as the best index of his well-being.

It is understood of course that the results of this paper apply only to human beings and are meant in no way to suggest desirable atmospheric conditions for the safe-keeping of merchandise, furniture, and fixtures of buildings, or for manufacturing processes.

A great deal of the discussion has centered around the comfort effect of a given effective temperature with wide differences in relative humidity. This discussion has included a consideration of the general comfort effect on occupants and also the shock effect upon entering or leaving. Taking up first the general comfort effect, we have to admit that we know relatively little about the subject. A most fruitful study which has been urged upon the Laboratory ever since the first comfort chart was presented is a study of relative humidities, as such, quite apart from their effect in determining a person's feeling of warmth or a person's comfort and well-being. That study has never been made. While this study has included some variations in the relative humidity of the atmospheric conditions at a given effective temperature, primary attention has been given to the effect on feeling of warmth and shock.

The discussion has directed a good deal of attention to the relative shock effect experienced by a person entering or leaving a space cooled to a given effective temperature at low or high relative humidities. This discussion has not been entirely consistent, and some have indicated that a greater cold shock should be experienced upon entering such a condition at low relative humidities, while others have indicated that a greater shock should be experienced with high relative humidities. Superficially there is some logic in both points of view, for with the higher relative humidity a lower dry-bulb temperature is maintained, while with the lower relative humidity the air is drier and may, superficially at least, be expected to result in excessive heat loss upon entering due to flash evaporation of perspiration on the body. Of the two, there is more reason, I believe, to expect a greater shock to result from flash evaporation at the lower humidity.

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As pointed out by a number of people discussing the paper, a person entering a cooled space from the hot outside will be perspiring so as to be in equilibrium with the outside condition. Upon entering the cooled space he does not need the availability of as much perspiration in order to provide the necessary heat loss by evaporation. Hence, it is obvious that more evaporation, and therefore more heat loss, will take place until the perspiration on the body has become normal for the inside condition. Hence, upon entering a cooled space there is actually an abnormally high total heat loss by radiation, convection and radiation. The cold shock is therefore actually the result of increased physical heat loss and not necessarily due to a feeling of contrast.

In discussing this phase of the subject, the drying of a dish cloth hung on a line has been mentioned as simulating the evaporation from a person, and of course, at the same effective temperature a dish cloth hanging on the line will dry or evaporate

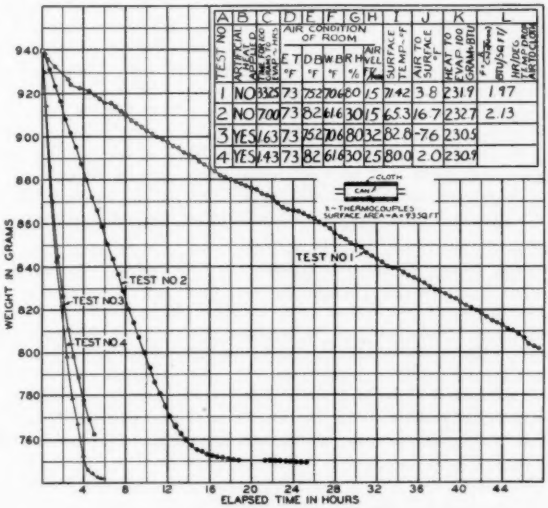

FIG. A. RESULTS OF FOUR TESTS

moisture faster in a low humidity than in a high humidity. We can prove, however, that the drying of this dish cloth does not simulate evaporation from the clothed human body. In order to dry, the dish cloth must obtain all of the heat for evaporation from the air. To evaporate moisture from the clothed body heat is supplied by the body from within as well as being supplied from the air. Since some of the discussion of this paper has accumulated we have succeeded in making a few brief tests which prove conclusively that the heat supplied from within the body is a much more dominant factor in evaporation of perspiration from the surface of the clothed body than the heat supplied from the atmosphere.

Referring to Fig. A, a sheet metal can, 3 in. in diameter and 12 in. long, with outlets at the two ends through which heated air could be circulated, was taken to simulate the clothed body. It was *clothed* with about ⅛-in. thickness of cheesecloth. Thermocouples gave the surface temperature of the metal under the *clothing*. These were actually soldered to the inside surface of the can. Other thermocouples were attached to the outside surface of the *clothing*. In the *first test*, this can without

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the supplying of heat to its inside was suspended in an atmospheric condition of 73.0 ET (75.2 F dry-bulb, 70.6 F wet-bulb, or 80 per cent relative humidity), in one of our psychrometric rooms, with the *clothing* evenly and fairly completely saturated with water. Weighings at approximately ½-min intervals gave the weight loss or evaporation curve. The *second test* was similar to the first, excepting that it was made in an atmospheric condition of 73.0 ET (82.0 F dry-bulb, 61.6 F wet-bulb, or 30 per cent relative humidity). It will be noted that with the lower relative humidity at the same effective temperature, the can without heat supplied to the inside lost heat very much more rapidly at the lower relative humidity. These two tests represent drying of a dish cloth in the air, or drying under conditions where no heat is supplied from within and where all of the heat of vaporization must come from the atmosphere. The *third test* was made under conditions similar to the first, excepting that heated air was passed through the can so as to maintain the can's metal surface temperature at 98.0 F. A *fourth test* was made under similar conditions, but in an atmospheric condition the same as in the second test. While the high and low relative humidity has some effect in the rate of drying, this effect is insignificant compared to the effect of the heat supplied within. The can under tests 3 and 4 simulated fairly well the clothed human body supplied with heat from within. It will be seen, therefore, that the clothed human body is relatively little affected in its rate of evaporation of moisture from its surface by rather wide variation in the relative humidity of the atmosphere, and this accounts for the facts established in our studies on the human subjects, namely, that there was no measurable difference in the rate of evaporation, and as a result no measurable difference in cold shock, of persons entering an atmosphere of a given effective temperature with either high or low relative humidities.

The results of the four tests plotted in Fig. A have been interesting to a number of students of the subject of summer comfort cooling and air conditioning, and also to students of drying. In this connection, it is particularly interesting to note that the rate of evaporation is entirely a heat transfer phenomenon. The can unheated from the inside must receive its heat by transfer from the air due to the temperature difference between the dry-bulb temperature of the air and its wet-bulb, or the temperature of the wet surface of the *clothed* can. It is of particular interest to note from the calculations contained in the figure that this results in a film conductance coefficient of 1.56 in the first test, and 2.04 in the second test, almost completely in agreement with the studies of film conductance coefficients for walls made by Rowley at Minnesota and by the Research Laboratory. On the other hand, when the can was heated from within the control factor was the transfer of heat from the inside of the can to the surface of the *clothing*. Further tests along this line have been suggested as of value in our knowledge of evaporative drying.

The authors are in complete agreement with many who discussed the paper that the paper should not be accepted as entirely conclusive in its application to all conditions of summer cooling and air conditioning. Further studies should be made of various phases of the subject, including variations in the desired effective temperature in summer cooled space in different geographical sections of the country under different climatic conditions; further studies should be made of the reactions of persons entering a cooled space for less than ½-hour occupancy, where the shock effect may be of greater significance than final equilibrium with the condition. Also, the study has again emphasized the desirability of an exhaustive study of the effect of relative humidity of the atmospheric condition in both winter and summer air conditioning, other than its effect on the feeling of warmth. This is, undoubtedly, one of the most fertile fields of investigation before us.

No. 1056

HEAT LOSSES AND EFFICIENCIES OF FUELS IN RESIDENTIAL HEATING

By R. A. SHERMAN * (*MEMBER*) AND R. C. CROSS ** (*NON-MEMBER*),
COLUMBUS, OHIO

OF the several factors that govern the choice of a fuel or the furnace or boiler in which it is burned for residential heating, the efficiency is by no means the most important. Heating satisfaction, which includes dependability, convenience, cleanliness, and quietness, is generally the determining factor that governs the choice of a fuel. If there is a choice between fuels, or if it is desired to weigh carefully the merits of the fuels in relation to their costs, efficiency should be considered, for it is not the cost of the heat in the fuel purchased but the cost of the heat delivered to the residence that should be compared.

Many laboratory determinations of the efficiency of various fuels in different types of heating equipment have been made. These have been valuable for the determination of the optimum values that can be obtained under correct and carefully controlled test conditions, but the possibility has always existed that the data were not applicable to actual residential operating conditions. Fuel tests conducted at the University of Illinois in the Research Residence of the *National Warm Air Heating and Air Conditioning Association* resulted in the recording of much valuable data, but in these tests the equipment was correctly installed; it was maintained in first class condition, and the attention to the fire was under technical control. Another valuable series of tests on coal firing was that conducted by the Bureau of Mines and published in a previous bulletin.[1] However, in these tests the equipment and conduct of the investigations were also under technical direction, and as a result, the performance obtained was probably better than that of customary operation.

A distinct need has existed, therefore, for data on the extent of the heat losses in the use of various fuels for residential heating. This need was partly filled by the extensive survey [2] of the performance of oil burners in residences in Madison, Wis. The purpose of the present investigation, which was spon-

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[1] Five Hundred Tests of Various Coals in House-Heating Boilers, P. Nicholls, S. B. Flagg and C. E. Augustine, Bureau of Mines Bulletin 276, 1928.

[2] Oil Burning in Residences, D. W. Nelson, A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 355.

Presented at the 43rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, St. Louis, Mo., January, 1937, by R. C. Cross.

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sored by Bituminous Coal Research, Inc., was to obtain this same type of information for various types of fuels and various types of heating equipment.

SCOPE OF INVESTIGATION

The investigation consisted of a series of tests on the heating plants of 15 homes located in Columbus, Ohio, during the 1935-36 heating season. The fuels included bituminous and semi-bituminous coals, by-product coke, natural gas, and oil burned in warm-air furnaces and hot-water and steam boilers.

A few of the houses included in the test program are illustrated in Fig. 1. They were homes of 7 to 14 rooms, occupied by families of 2 to 7 persons, and considered to be typical of homes of this class in any part of the country.

METHOD OF INVESTIGATION

Test Procedure

The test on each heating plant was normally of 123 hours' duration, from 6 a.m. Monday to 9 a.m. Saturday, during which period observers were present from 6 a.m. to 10 p.m. The data recorded included the weight of fuel charged and of ash and refuse removed, the frequency and time of attention to the plant, the temperature of the air and of the flue gases, the composition of the flue gases, density of smoke, and the drafts. A recorder in the living quarters of the house recorded the temperature, humidity, and the operating intervals of stokers and oil burners. A watt-hour meter was used to determine the power consumption of stokers and oil burners. A gas meter was connected into the line to the furnace on gas-fired installations to obtain the amount of gas used.

An installation of the test equipment on a warm-air furnace is shown in Fig. 2. The apparatus was made compact and portable so that it could readily be moved from one house to another.

An attempt was made in these tests to obtain normal operation of the heating equipment so that the results would be representative of normal practice. The householder fired the furnace, adjusted the dampers, set the thermostat in accordance with his comfort standards, and the observer merely recorded the results obtained. At the end of each test the householder was advised of changes which could be made in his firing practice or adjustment of the stoker or burner to improve the results.

Calculation of Efficiencies

When a boiler or furnace is tested in the laboratory the amount of heat utilized is usually determined directly. That is, in a hot-water boiler, the weight of water circulated through the boiler and its rise in temperature are measured; in a steam boiler, the weight of water evaporated and the temperature and pressure of the steam are determined; in a warm-air furnace, the weight of air and its rise in temperature are found. These methods were not available in the test houses. Therefore, it was necessary to determine the amount of the various losses of heat and obtain the efficiency by difference.

FIG. 1. TYPICAL HOUSES USED IN TESTS

The heat losses in domestic heating that can be measured are: (1) loss in the moisture from the air, in the fuel, and that formed from the combustion of hydrogen in the fuel, (2) the sensible heat in the dry flue gases, (3) the undeveloped heat in carbon monoxide, and (4) heat in unburned carbon in the ashpit. In laboratory tests the radiation from the boiler or furnace is con-

sidered as a loss, but in these tests it was considered as useful heat as it was delivered to the house.

In addition to these losses, however, there is always an item of *unaccounted-for loss,* which is generally due to unburned hydrocarbons and soot which are not ordinarily determinable. Inasmuch as this loss is often an appreciable fraction of the heat input, it was obvious that the efficiencies could not be taken as the difference between the heat input and the determined losses. From several available sources values for *unaccounted-for losses* were selected and added to the determined losses before calculation of the heat utilized, or efficiency, by difference. The values assumed and the basis for their selection are discussed under each group in the presentation of the results.

FIG. 2. ARRANGEMENT OF TEST EQUIPMENT

Although the values obtainable by these methods cannot be considered to have the accuracy of tests under laboratory conditions, and although it was impossible to cover enough of each type to arrive at true averages, yet the fact that the data obtained were from truly typically maintained and operated home heating plants is believed to outweigh the limitations in accuracy.

DISCUSSION OF DATA

Tables 1 to 4 present the principal data obtained from the tests of the 15 heating plants; each table includes the tests of one method of firing. The data are much abridged from the total data taken, but the most significant data are given and many are included that are not discussed, which may be of additional interest or value. The general indications are summarized at the end of the discussion of each fuel and method of firing, and the overall conclusions are summarized at the close of the paper.

Bituminous Coal, Hand-Fired

Table 1 presents the results of the tests of the three heating plants which burned bituminous coal fired by hand; two were warm-air furnaces and the third a steam boiler. All three were typical installations of their kind with the exception that all were equipped with thermostatic controls; that of Case 32 had a *high-limit* control based on the expansion of a copper pipe in the bonnet and the steam boiler had the usual limit control actuated by the boiler pressure, but the furnace of Case 13 had no limit control.

Both Case 13 and Case 26 used low-ash, lump coal, whereas Case 32 used 2 in. screenings which had a higher ash content.

The average CO_2 content of the flue gases was similar for Cases 13 and 26. However, the weight of dry flue gases per pound of coal, from which the dry flue gas loss is calculated, was higher for Case 26; this apparent discrepancy results from the fact that the gas composition reported is the arithmetical average for the week, whereas the weight of the flue gas was calculated for each hourly period and then averaged, which is the correct method. The CO_2 content for Case 32 was much lower than the other two. This was caused by the admission of air through three pipes which the owner had placed in the furnace to admit air over the fuel bed. The air admitted may not have been excessive in periods of high rates of burning, but during the test week, which was in rather mild weather, too much secondary air was admitted.

In all three installations some unburned *CO* was found in the flue gases. This accounted for a loss of 1.3 to 4.0 per cent of the heat in the coal fired.

The losses due to unburned carbon in the ash were both very favorably low for Cases 13 and 26 but were higher for Case 32. This was accounted for by the small size of the coal fired and a broken grate in the furnace.

The loss of heat in the dry flue gases, which is the largest single item of heat loss other than the *unaccounted-for* was 16.3 to 27.8 per cent. The higher loss for Case 26 is accounted for by the higher weight of flue gases and the higher temperature. A higher temperature was expected because of the more severe weather during the test on this installation. The high loss in Case 32 was due to the high excess of air.

Unaccounted-for Loss

The necessity for the assumption of some value for an *unaccounted-for* item which includes the undetermined losses in soot, tar, and gaseous hydrocarbons has been previously discussed. For each of these tests a value of 20 per cent has been assumed. A series of tests conducted by the Bureau of Mines on a small boiler in which the radiation was separately measured and in which the entire installation was mounted on scales to permit accurate knowledge of the amount of fuel burned gives the most authoritative data on the losses to this source. Papers [3-4] on this work indicated losses of 15 to 25 per cent which

[3] Heat Transference and Combustion Tests in Small Domestic Boiler, John Blizard, W. M. Myler, Jr., J. K. Seabright and C. P. Yagloglou, A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923.
[4] Heat Transference and Combustion Tests in Small Domestic Boiler, H. W. Brooks, M. L. Orr, W. M. Myler, Jr., and C. A. Herbert, A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925.

XUM

TABLE 1—PERFORMANCE DATA

HAND-FIRED PLANTS

ITEM No.		GROUP 1 BITUMINOUS COAL			GROUP 2 SEMI-BITUMINOUS COAL		GROUP 3 COKE	
	GENERAL INFORMATION							
1	Case number	13	26	32	14	4	1	Wood
2	Type of house	Brick	Brick	Wood	Wood	Brick	Brick	9
3	Number of rooms	7	9	8	7	8	8	2
4	Maximum design heat loss of building—Btu per hr	71,156	95,319	69,589	88,816	84,924	83,261	77,807
5	Type of heating system	Warm Air	Steam	Warm Air	Warm Air	Warm Air	Warm Air	Warm Air
6	Leader area—sq in	565	...	712	770	623	772	791
7	Radiation—sq ft	...	448	...	...	...	...	...
8	Date of test	Nov. 18–23, 1935	Feb. 17–22, 1936	Mar. 9–14, 1936	Jan. 6–11, 1936	Feb. 10–15, 1936	Oct. 28–Nov. 2, '35	Jan. 20–25, 1936
9	Duration of test—hrs	121.5	123	122	122	123	120	122
10	Degree-days in test period	137	266	113	150	219	29	294
11	Average daily mean temperature—F	36	14	43	36	21	59	8
	FUEL							
12	Kind	Eastern Ky.	W. Va.	Ohio	Semi-bit.	Semi-bit.	Coke	Coke
13	Size	Lump	Lump	2″ screenings	Lump	Lump	Nut	Stove & Nut
	Proximate analysis, as fired, per cent:							
14	Volatile matter	33.4	32.0	33.6	23.7	15.7	1.1	1.2
15	Fixed carbon	60.0	63.0	50.9	71.2	74.4	90.3	90.3
16	Ash	3.6	3.5	7.5	3.1	6.6	7.8	7.7
17	Moisture	3.0	1.5	8.0	2.0	3.3	0.8	0.8
18	Sulphur	0.6	0.5	2.2	0.6	0.7	0.6	0.7
19	Calorific value, Btu per lb	14,143	14,681	11,928	14,942	14,313	12,395	12,332
20	Total fuel-fired—lb	429	867.25	440.25	425.5	667	171.25	676.5
	DRAFTS							
21	Draft in furnace—in. water	0.016	0.060	0.071	0.025	0.069	0.025	0.018
22	Draft at smokehood—in. water	0.016	0.046	0.068	0.042	0.074	0.017	0.013

	FLUE GASES							
23	CO_2—per cent	7.8	7.8	4.0	6.0	5.7	6.4	10.3
24	O_2—per cent	11.2	11.2	16.0	13.7	13.9	13.7	9.9
25	CO—per cent	0.6	0.2	0.1	0.1	0.1	0.3	0.3
26	N_2—per cent	80.4	80.8	79.9	80.2	80.3	79.6	79.5
27	Dry flue gas per lb of fuel—lb	26.18	33.27	47.75	37.44	36.78	33.7	21.52
28	Excess air—per cent	105	108.5	310	182	188.5	182	86.7
	TEMPERATURES							
29	Flue gases—F	454	574	363	428	463	219	475
30	House living quarters—F	73	74	71	68	72	72	68
31	House living quarters—relative humidity—per cent	40	28	50	43	26	58	30
	HOURLY RATES							
32	Fuel-fired—lb per hr	3.53	7.04	3.61	3.54	5.43	1.43	5.54
33	Heat input—Btu per hr	49,925	103,354	43,060	52,895	77,719	17,725	63,319
34	Fuel-fired—lb per sq ft grate per hr	1.33	1.65	1.25	1.47	2.05	.38	1.39
	SMOKE							
35	Average density—Ringelmann number	0 to 1	1	0	0	0	0	0
	ATTENDANCE							
36	Total number times attention to fire per day	5	6	4	11	4		5
	HEAT BALANCE—PER CENT							
37	Losses due to moisture	4.7	4.6	5.9	4.0	4.0	0.7	0.7
38	Loss due to heat in dry flue gases	16.8	27.8	27.6	21.5	24.4	9.6	17.0
39	Loss due to carbon monoxide	4.0	1.4	1.3	0.9	1.0	3.2	2.0
40	Loss due to unconsumed carbon in ash	1.0	0.4	3.8	4.6	3.3	0.6	0.5
41	Unaccounted losses (assumed)	20.0	20.0	20.0	15.0	15.0	10.0	5.0
42	Total losses	46.5	54.2	58.6	46.0	47.7	24.1	25.2
43	Overall efficiency (by difference)	53.5	45.8	41.4	54.0	52.3	75.9	74.8
44	Total	100.0	100.0	100.0	100.0	100.0	100.0	100.0

were unaccounted for. A bulletin[5] of the Mines Branch, Department of Mines, Canada, showed that unaccounted-for losses increased with the increase in the volatile matter of the coal; with coals having 30 to 35 per cent volatile matter the loss averaged 15 per cent.

The tabulation given shows a comparison of the heat balance of Case 26 with that of one of the tests† previously reported:

HEAT BALANCE, PER CENT OF HEAT IN COAL	CASE 26	B. M. TEST 52[a]
Heat utilized	45.8	53.5[b]
Losses due to moisture	4.6	4.2
Losses to heat in dry flue gases	27.8	13.5
Loss due to carbon monoxide	1.4	4.2
Loss to unburned carbon in ash	0.4	4.4
Unaccounted-for loss	20.0	20.2
	100.0	100.0
CO_2 in flue gases, average, per cent	7.8	7.1
Flue gas temperature, degree Fahrenheit	574.0	406.0
Volatile matter in coal, per cent	32.0	36.4
Fuel fired per square foot per hour	1.3	1.3

[a] Calculated to include combustible in ash loss.
[b] Includes radiation and convection losses.

These data indicate that the assumed value of 20 per cent for the unaccounted-for losses was a reasonably accurate approximation.

Summing up the various losses and deducting from 100, there remains 41 to 53 per cent of the heat in the fuel as the overall thermal efficiency or the heat delivered to the house.

Semi-Bituminous Coal, Hand-Fired

Two tests were recorded on homes using semi-bituminous coal in lump size, one a medium-volatile, 23.7 per cent, and one a low-volatile, 15.7 per cent, coal. Both installations were warm-air furnaces. Neither furnace was equipped with a thermostat and the fire was attended and the dampers adjusted according to the desire of the occupants. The installations were typical, therefore, of heating plants with no automatic devices. Table 1 includes the data on these tests.

The maximum CO_2 content of the flue gases in any hour on one case was 12.4 and on the other 10.7; the minimum was as low as 1.0 per cent. The average CO_2 content for both furnaces was about 6.0 per cent, which amounts to about 180 per cent excess air. The flue-gas temperatures were similar, and the loss of heat in dry flue gases was of the same order, 21.5 and 24.4 per cent for the two installations. The loss due to moisture was the same, 4.0 per cent, which is a normal loss. The loss due to unburned carbon monoxide was about

[5] Comparative Tests of Various Fuels When Burned in a Domestic Hot Water Boiler, E. S. Malloch, and C. E. Baltzer, Canada Department of Mines, Mines Branch, 705, 1929.
† Loc. cit. See note 3.

1.0 per cent, which was less than for the furnaces burning the high-volatile coals.

The losses due to the unburned carbon in the ashpit were 4.6 and 3.3 per cent, which were higher than those for the high-volatile coals. This is explained by the greater friability of the lower volatile coals.

The value selected for the losses in tar, soot, and gaseous hydrocarbons, the *unaccounted-for loss,* for these coals was 15.0 per cent. This value was chosen from a consideration of previously reported data [6] where the average *unac-*

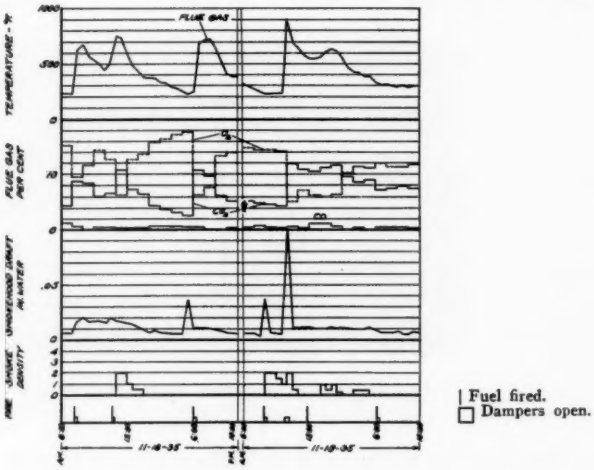

FIG. 3. TEST LOG, CASE 13, B-WA-BH

counted-for loss, including radiation, for coals of 15 to 25 per cent volatile matter was 13.25 per cent.

The sum of the losses amounts to 46.0 and 47.7 per cent, leaving as the efficiency of utilization of the heat in the coal 54.0 and 52.3 per cent. These values check closely with the result with high-volatile coal, Case 13, which had a similar flue-gas temperature; the lower *unaccounted-for loss* assumed for the low-volatile coals was offset by the lower loss in heat in dry flue gases because of the lower excess air with the high-volatile coal.

Coke, Hand-Fired

The results of the two tests operated with furnaces burning hand-fired coke in warm-air furnaces in houses of similar size are also included in Table 1. The test on Case 1 was recorded during a week of mild weather in early fall

[6] Loc. cit. See note 5.

and that on Case 9 during one of the most severe weeks of the winter. The furnace of Case 1 was arranged with a fan and filter unit, and both furnaces were equipped with thermostatic control and limit controls to prevent overheating of the furnace. Both installations were somewhat above the average in the order of maintenance. Also both owners had used coke for a number of years and fired it with care.

The coke used in both furnaces was by-product coke from the same source, but Case 1 used nut size whereas Case 9 used a mixture of stove and nut. The coke had a lower ash content than that frequently sold for domestic purposes; for the country as a whole the ash content of coke would probably average 9 to 10 per cent, whereas this coke contained less than 8 per cent ash.

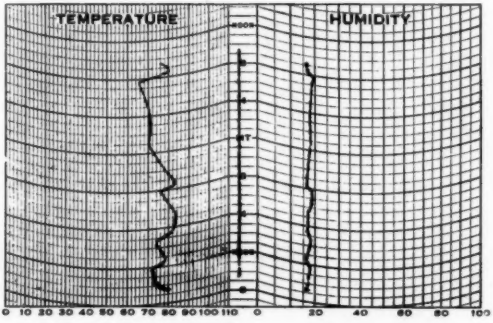

FIG. 4. HOUSE TEMPERATURE RECORD—MANUAL CONTROL

Because of the difference in the average rates of burning, 0.38 and 1.39 lb per square foot of grate per hour, the average CO_2 contents were 6.4 and 10.3 per cent respectively. During the test on Case 9 an hourly average as high as 16.4 per cent CO_2 was found. The higher weight of flue gas per pound of fuel in Case 1 was more than compensated for by the low flue gas temperature, 219 F, so that the loss in sensible heat in the dry flue gas was 9.6 per cent as compared to 17.0 per cent for Case 9.

Because of the low content of moisture and hydrogen in the coke, the moisture losses were low, below 1 per cent, and because of the careful attention used in firing the losses due to unburned carbon in the ash were low. With both installations an average of 0.3 per cent CO was found in the gases which accounted for losses of 3.2 to 2.0 per cent.

Although no loss in tar, soot, or gaseous hydrocarbons is expected with coke as a fuel, some allowance must be made for *unaccounted-for losses*. A consideration of previously recorded data [7] which included tests on coke showed that at the lower rate of combustion, Case 1, a value of 10.0 per cent, and at the higher rate, Case 9, a value of 5.0 per cent would be close approximations to the correct value.

[7] Loc. cit. See note 3.

The thermal efficiencies calculated by difference after summing up the losses and using the estimated *unaccounted-for losses* amount to 75.9 and 74.8 per cent for the two cases.

GENERAL SUMMARY OF HAND-FIRED TESTS

Cyclic Operation. In Fig. 3 is a log of the test results recorded for two days of operation on Case 13, which burned eastern Kentucky lump coal in a warm-air furnace. The cycle of operation, that is, the increase of the smoke, of the CO_2 in the flue gases, and of the flue gas temperature after firing, followed by a gradual drop until the next firing period is typical of the results in other

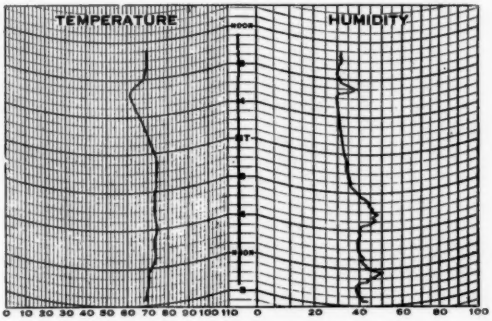

FIG. 5. HOUSE TEMPERATURE RECORD—THERMOSTATIC CONTROL

hand-fired tests. Although this furnace had a damper motor controlled by a room thermostat, it had no high-limit control, as has been previously mentioned. With a high-limit control to prevent the overshooting of the temperature of the furnace, the range of variations in the cycles can be materially reduced.

Effect of Thermostatic Control. A 24 hour record of the room temperature maintained in Case 4 is shown in Fig 4 where the fire was controlled manually, and in Fig. 5 is a similar record for Case 26 where the dampers were controlled by a room thermostat supplemented by the high-limit control from the boiler pressure. The value of the thermostatic control in the maintenance of more uniform temperatures even with hand-firing is clearly shown by these records.

Smoke. Smoke intensities from the stacks of these furnaces varied considerably. Except in Case 26, high-volatile coal burned in a steam boiler, where readings as high as No. 4 Ringelmann chart were noted and the stack was seldom entirely clear, the smoke intensities and periods of smoke duration were not excessive. The amount of smoke was less with the semi-bituminous coals, and obviously none was found when burning coke.

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TABLE 2—PERFORMANCE DATA

STOKER-FIRED PLANTS

ITEM NO.		*On* PERIOD	*Off* PERIOD	TOTAL PERIOD	*On* PERIOD	*Off* PERIOD	TOTAL PERIOD	*On* PERIOD	*Off* PERIOD	TOTAL PERIOD
	GENERAL INFORMATION									
1	Case number		2			5			10	
2	Type of house		Brick			Brick			Wood	
3	Number of rooms		10			11			12	
4	Maximum design heat loss of building—Btu per hr		107,324			113,234			76,655	
5	Type of heating system		Steam			Hot Water			Warm Air	
6	Leader area—sq in.		...			...			846	
7	Radiation—sq ft		686			700			...	
8	Date of test		Nov. 5-10, 1935			Dec. 2-7, 1935			Jan. 27-31, 1936	
9	Duration of test—hrs.		122.5			123			123	
10	Degree-days in test period		90			194			297	
11	Average daily mean temperature —F		46			28			8	
	FUEL									
12	Kind		Eastern Kentucky			West Virginia			West Virginia	
13	Size		1½″ x ⅜″			1½″ x ⅜″			1½″ x ⅜″	
	Proximate analysis, as fired, per cent									
14	Volatile matter		33.9			35.3			36.7	
15	Fixed carbon		56.7			54.6			56.8	
16	Ash		4.9			6.8			3.9	
17	Moisture		4.5			3.3			2.6	
18	Sulphur		0.7			2.1			0.7	
19	Calorific value, Btu per lb		13,407			13,615			14,160	
20	Total fuel fired—lb	....	....	348.5	....	....	1038.25	....	....	202.5
	DRAFTS AND PRESSURES									
21	Stoker windbox pressure—in. water	0.099	0.054	....	0.564	0.032	....	0.990	0.0	....
22	Draft in furnace—in. water	0.086	0.067	....	0.029	0.041	....	0.164	0.155	....
23	Draft at smokehood—in. water	0.088	0.063	....	0.026	0.031	....	0.189	0.180	....

	FLUE GASES									
24	CO_2—per cent	6.9	3.5	5.3	9.4	7.7	9.3	7.8	6.4	6.9
25	O_2—per cent	12.2	16.5	14.3	9.1	10.6	9.4	11.3	13.6	12.4
26	CO—per cent	0.3	0.0	0.0	0.2	0.2	0.2	0.0	0.0	0.0
27	N_2—per cent	80.6	80.0	80.4	81.3	81.5	81.1	80.9	81.0	81.7
28	Dry flue gas per lb. of fuel—lb	30.0	55.1	41.8	21.9	25.8	22.4	27.0	33.3	30.1
29	Excess air—per cent	130.7	357.0	206.5	72.3	95.5	76.8	113.0	174.7	135.8
	TEMPERATURES									
30	Flue gases—F	437	268	358	934	430	882	786	720	749
31	House living quarters—F	....	....	72	....	....	75	....	....	72
32	House living quarters—relative humidity—per cent	....	....	51	....	....	22	....	....	26
	HOURLY RATES									
33	Fuel fired (average) per hr	....	....	2.84	....	....	8.44	....	....	9.78
34	Heat input (average)—Btu per hr	....	....	38,076	....	....	114,911	....	....	138,485
	SMOKE									
35	Average density—Ringelmann number	0	0 to 4	....	....	....	....	0 to 1	0 to 2	....
	ATTENDANCE									
36	Total number times attention to stoker per day	....	....	4	....	....	2	....	....	2
	STOKER OPERATION									
37	Average length *On* period—Min.	7.8	....	....	29.5	....	....	22.8	....	....
38	Average length *Off* period—Min.	....	47.4	....	....	36.6	....	....	87.0	....
	POWER									
39	Total power—kwhr	....	....	2.7	....	....	9.2	....	....	11.3
40	Power per ton of coal-fired—kwhr	....	....	15.5	....	....	17.7	....	....	18.8
	HEAT BALANCE—PER CENT									
41	Losses due to moisture	....	....	4.9	....	....	5.4	....	....	5.0
42	Loss due to heat in dry flue gases	....	....	21.1	....	....	32.3	....	....	34.7
43	Loss due to carbon monoxide	....	....	0.0	....	....	1.2	....	....	0.0
44	Loss due to unconsumed carbon in ash	....	....	0.0	....	....	0.0	....	....	0.3
45	Unaccounted losses (assumed)	....	....	5.0	....	....	5.0	....	....	5.0
46	Total losses	....	....	31.0	....	....	43.9	....	....	45.0
47	Overall efficiency (by difference)	....	....	69.0	....	....	56.1	....	....	55.0
48	Total	....	....	100.0	....	....	100.0	....	....	100.0

Attendance to Fire. Of the five tests in homes burning hand-fired coal, the number of trips to the basement to attend to the fire, either charging fuel, shaking grate, or stirring up the fuel bed, varied from 4 to 6, except for Case 14 where attention was given on an average of 11 times daily, including 6 times to charge fuel. As the weather was not severe during the week, this amount of attention should have been unnecessary and was due to lack of knowledge of proper methods of firing.

The attendance to the fire in the cases using coke averaged twice daily for Case 1 during the week of mild weather and 5 times daily for Case 9 during the week of very severe weather. The conclusion may be drawn, therefore, that when properly fired the attention to a coke fire may be somewhat less than to a furnace burning hand-fired bituminous coal.

BITUMINOUS COAL, STOKER-FIRED

The results of the tests of the three heating plants in which bituminous coals were fired by stokers are presented in Table 2; one was a steam boiler, one was a hot-water boiler, and the third was a warm-air furnace. The stokers of Cases 2 and 5 were typical of the common screw-feed type underfeed stokers and were designed for removal of ash as clinker; these were installed in 1932 and 1933, respectively. The stoker of Case 10, which was installed in 1929, differed from the conventional design in that it had a rotary grate around the retort which was moved by hand, thus removing a part of the ash to an ashpit as fine ash. The controls of Cases 2 and 5 included a room thermostat, a high-limit control, and a clock-driven hold-fire control. No high-limit control was provided with the stoker of Case 10.

The coals used were sized *stoker coals,* 1½x⅜ in. That used in Case 2 was an eastern Kentucky coal, and the other two used oil-treated West Virginia coal.

The flue gas from these installations was sampled continuously throughout the period of observations, but the samples were divided into the *off* and the *on* periods of operation, as previous experience had indicated that the composition differed markedly in the two periods. To obtain an overall average of the gas composition for purposes of calculation, it was necessary to have some measure of the relative amounts of coal burned in the two periods. No direct measure was had, but an approximation was made by the assumption that the rate of burning was proportional to the square root of the pressure drop across the fuel bed. With this value and the relative time of operation, the weighted averages of the gas composition and the flue-gas temperature were obtained.

The boiler of Case 2 had a rectangular fire box in which the stoker retort was installed near the front; the feed tube entered diagonally from the side of the boiler near the rear. Due to the characteristics of the coal or the stoker, or both, the coal fed principally to the rear of the retort, often leaving bare tuyeres at the front. The coal tended to coke into spires which would fall away from the retort; unless they were manually pulled back into the retort at intervals, they would not burn. The operator made a practice of leveling the fire in this way two to three times a day.

The weather during the week of the test on this installation was very mild, and the ratio of *on* to *off* operation was about 1 to 6, the stoker operating principally as demanded by the hourly operation of the hold-fire control. Although the CO_2 averaged about 7 per cent during the *on* periods, owing to the long *off* periods and the fact that the furnace draft was higher than it should have been, the overall average was only 5.3 per cent, or over 200 per cent excess air. Although a small amount of *CO* was found during *on* periods of operation, the average was so low that no loss resulted from this cause. No combustible was found in the refuse which was removed as clinker, and, therefore, no loss resulted. The boiler had a comparatively large heat-absorbing surface, and the flue-gas temperatures were, therefore, favorably low despite the high excess of air. The loss due to sensible heat in the dry flue gases was 21.1 per cent.

Because of the proximity of the surrounding surfaces in the round boiler of Case 5, any coke spires that were formed fell back into the fire, and without attention to the fuel bed other than at the times of removing clinker the fuel bed remained in good condition. This is evidenced by the higher CO_2 content of the flue gases in both the *on* and *off* periods. As the ratio of the periods, because of the colder weather and the characteristics of the plant, was about 1 to 1, and the furnace draft was maintained at a low value, the overall average CO_2 was 9.3, which was equivalent to about 77 per cent excess air.

No loss was found in combustible in the refuse, and only a small amount because of carbon monoxide in the flue gases, but the performance of the boiler and stoker was limited owing to the high flue-gas temperatures. These averaged 934 F for the *on* periods and for the entire test 882 F. The boiler had only one intermediate section; if it had had more intermediate sections, much more favorable efficiency would have been obtained.

The stoker of Case 10 operated differently from the other two in that combustion must have taken place from air entering through the perforated grate around the retort as well as through the retort by natural draft during the *off* periods. The furnace draft during the *off* periods averaged 0.155 in. of water, which is about three times that commonly accepted as good practice, and the ratio of *on* to *off* time was 1 to 3 as compared to 1 to 1 for Case 5, although the weather during the test period was much more severe.

The average CO_2 for the entire test was 6.9, corresponding to 136 per cent excess air. No *CO* was found, and the loss due to combustible in the ashpit was less than 1 per cent. The flue-gas temperatures were much higher in both the *off* and the *on* periods than they should have been, which shows that the heat-absorbing surface of the furnace was too small. This resulted in a loss due to sensible heat in the dry flue gases of about 35 per cent.

No published heat balances of small stokers in domestic boilers are available from which information could be obtained on the expected *unaccounted-for loss.* However, data from tests of stokers of similar nature now being conducted in the laboratory of the Institute indicate that 5 per cent of the heat in the coal might be lost in such unaccountable items as fly carbon, soot, and hydrocarbons, and this value was assumed.

The overall efficiencies were 69 per cent for the boiler tested during the extremely mild weather and 55 and 56 per cent for the installations tested during the colder weather, when the various losses were deducted from 100.

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GENERAL SUMMARY OF STOKER-FIRED TESTS

Relation of Stoker to Boiler or Furnace. Except for Case 2, the efficiencies obtained in these tests were less than have been assumed to be obtained with small stokers. Although the number tested was too small to draw any broad general conclusions that the performance of all others is of similar order, yet these three picked at random are doubtless typical of many installations.

The relatively good efficiency of Case 2 was due largely to the ample heat-absorbing surface of the boiler and was obtained at the expense of more frequent attention to the fire, to level it off, than should have been required. The stoker was definitely at fault in not giving uniform distribution of the fuel,

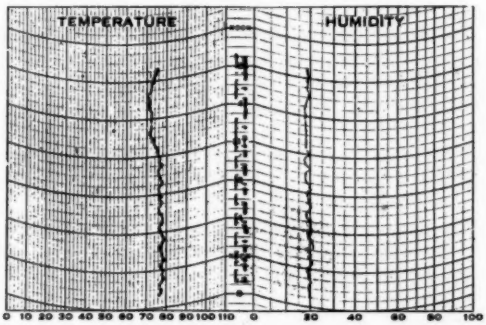

FIG. 6. HOUSE TEMPERATURE RECORD—STOKER INSTALLATION

and the installation was at fault in not so confining the hearth that the coke formations would necessarily fall back into the retort.

The low efficiencies of the other two installations were principally due to the lack of surface to absorb the heat efficiently at the rate at which it was liberated. Both the boiler and the furnace were designed primarily for hand-firing where the rates of combustion, except for short periods in the morning, are of the order of perhaps 6 to 10 lb of coal per hour. These stokers burned during *on* periods at 20 to 30 lb per hour, and the surfaces could not absorb the heat at that rate.

Previous reports [8] on stokers which burn anthracite have shown that efficiencies of 70 per cent could be obtained in a boiler designed for stoker use, whereas about 63 per cent was the maximum that could be obtained with the stoker installed in a boiler designed for hand-firing.

Smoke. Because of the open check draft on Case 5, used to reduce the furnace draft, any smoke produced was so diluted with air that no readings were taken. On Case 2, no smoke greater than No. 1 and on Case 10 none

[8] Performance Expectancy of Domestic Underfeed Stokers for Anthracite, Allen J. Johnson, Transactions, *A.I.M.E.*, Coal Division, Vol. 119, 1936.

greater than No. 2 Ringelmann chart was observed; this was much lighter than on the hand-fired tests of high-volatile coal.

Uniformity of Temperatures. A typical example of the type of regulation of room temperature obtained with the stoker on Case 5 is shown in Fig. 6. The temperature was held during the day to within 2 F. The marks in the center of the chart indicate times of operation of the stoker; the correlation between the temperature of the room and the operation of the stoker can readily be seen.

Attendance to Fire. The necessity for attention to the fire on Case 2, because of the improper design of the stoker hearth, has been mentioned. Attention was given the others but twice each day, to fill the hopper and remove clinker, during the test periods; once per day would normally have been enough. This is contrasted with the number of 4 to 11 times per day for the hand-fired installations.

NATURAL GAS

The results of the tests of the three heating plants which were fired with natural gas are presented in Table 3. Case 6 was a warm-air, furnace-burner unit, Case 33 was a boiler-burner unit, and Case 30 was a warm-air furnace with a conversion burner. All were equipped with room thermostats, and the boiler unit had a high-limit control actuated by the boiler pressure.

In the units of both Case 33 and Case 30 the air-gas ratio was well adjusted; the CO_2 content of the flue gases averaged 8.1 and 8.8 per cent, respectively, corresponding to 38 and 25 per cent excess air. In the unit of Case 6, however, although specifically designed for gas, a large opening around the burners had no controllable shutters, and so much air was admitted that the CO_2 was only 3.3 per cent. In all units the CO_2 remained practically constant during the entire test period when the burners were in operation.

As no unburned carbon monoxide was found in the flue gases and there was no ashpit loss, only three losses remain. The loss due to moisture was very nearly the same for the three installations. This is governed principally by the amount of hydrogen in the gas and amounted to 10 to 11 per cent, about twice that for coal-fired furnaces. The loss due to sensible heat in the dry flue gases was 8.3 to 15.6 per cent. The high loss for Case 6 was caused by the very high excess air, and that for Case 30 was caused by the high flue-gas temperature. The warm-air furnace, designed for coal, did not have the heating surface required to absorb the heat from the gases efficiently.

The *unaccounted-for loss* in firing by gas is due to errors and possibly to small amounts of undetermined combustible gases. In other tests [9] the average *unaccounted-for loss*, excluding radiation, for thirteen tests was 4.4 per cent. Therefore, a value of 5 per cent was assumed for these tests.

The deduction of the measured and assumed *unaccounted-for losses* from 100 left 69 to 76 per cent of the heat in the fuel as the overall thermal efficiency of these gas-fired units.

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[9] Loc. cit. See note 3.

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TABLE 3—PERFORMANCE DATA

GAS-FIRED PLANTS

ITEM NO.		*On* PERIOD	*Off* PERIOD	*On* PERIOD	*Off* PERIOD	*On* PERIOD	*Off* PERIOD
	GENERAL INFORMATION						
1	Case number	6		33		30	
2	Type of house	Wood		Wood		Brick	
3	Number of rooms	8		14		11	
4	Maximum Design heat loss of building—Btu per hr	79,988		87,978		72,412	
5	Type of heating system	Warm-Air		Steam		Warm-Air	
6	Leader area—sq in	646		...		772	
7	Radiation—sq ft	...		602		...	
8	Date of test	Dec. 9-13, 1935		Feb. 24-29, 1936		Mar. 2-6, 1936	
9	Duration of test—hrs	112		123		123	
10	Degree-days in test period	136		113		135	
11	Average daily mean temperature —F	36		43		38	
	FUEL						
12	Kind	Natural Gas		Natural Gas		Natural Gas	
13	Calorific value (gross)—Btu per cu ft	1,000		1,000		1,000	
14	Total gas burned—metered—cu ft	4,370		5,795		4,613	
15	Total gas burned—corrected to 30/60—cu ft	4,375		5,761		4,594	
	DRAFTS						
16	Draft in furnace—in. water	0.006	0.008	....	....	0.011	0.019
17	Draft at flue gas outlet—in. water	0.055	0.036	0.010	0.003	0.017	0.020

	FLUE GASES								
18	CO_2—per cent	3.3	....		8.1	....		8.8	....
19	O_2—per cent	14.4	....		6.2	....		4.6	....
20	CO—per cent	0.0	....		0.0	....		0.0	....
21	N_2—per cent	82.3	....		85.7	....		86.6	....
22	Dry flue gas per lb of fuel—lb	42.25	....		17.67	....		16.4	....
23	Excess air—per cent	196.8	....		37.8	....		25.2	....
	TEMPERATURES								
24	Flue gases—F	346	....		427	....		687	....
25	House living quarters, F		71			74			65
26	House living quarters—relative humidity—per cent		44			38			46
	HOURLY RATES								
27	Fuel burned (average) cu ft per hr		38.95			47.15			37.5
28	Heat Input (average)—Btu per hr		38,950			47,150			37,500
	BURNER OPERATION								
29	Average length *On* period—min.	12.0	....		25.2	....		66	....
30	Average length *Off* period—min.	....		35.4	....		104.3	....	194
	HEAT BALANCE—PER CENT								
31	Losses due to moisture	10.3	....		10.6	....		11.5	....
32	Loss due to heat in dry flue gas	15.6	....		8.3	....		13.4	....
33	Unaccounted losses (assumed)	5.0	....		5.0	....		5.0	....
34	Total losses	30.9	....		23.9	....		29.9	....
35	Overall efficiency (by difference)	69.1	....		76.1	....		70.1	....
36	Total	100.0	....		100.0	....		100.0	....

GENERAL SUMMARY OF GAS-FIRED TESTS

Relation of Equipment to Operation. The one boiler-burner unit tested gave an efficiency approaching that generally assumed to obtain with gas-fired units; also the conversion burner in the warm-air furnace gave an efficiency as high as could be expected, although the flue-gas temperature was high. The warm-air furnace designed for gas gave surprisingly low results because of the very high excess air. These data show the value of careful adjustment of the air-fuel ratio and also show that it is inadvisable to assume that all gas-fired burners operate with the publicized performance of the best.

OIL-FIRED TESTS

The results of tests of the heating plants which were fired with oil are presented in Table 4. Case 11 was a warm-air furnace which burned No. 3 oil with a pressure-atomizing burner, and Case 8 was a round cast-iron boiler which used No. 1 oil with a center-flame, rotary burner. Both installations were equipped with room thermostats and high-limit controls which gave good regulation of the house temperature.

The CO_2 content of the flue gases was similar in both installations, about 7 per cent, corresponding to about 100 per cent excess air. This is a lower CO_2 content than these burners should give under normal operation. The flue-gas temperature was not excessively high, 547 F, for the warm-air furnace but was much higher than it should have been, 866 F, for the steam boiler. The difference was partly, but not entirely, accounted for by the difference in weather during the week of the tests. Because of the low CO_2 the loss in heat in the dry flue gases was higher than is to be expected for oil-fired installations, and was particularly high, 30.3 per cent, for the steam boiler.

The loss due to moisture was about 8 per cent for both cases, higher than for coal, but somewhat less than for natural gas, as the hydrogen content of oil is intermediate between coal and gas.

In a series of tests [10] of oil-fired boilers an average value for 15 tests of 14.7 per cent for *Radiation and Unaccounted-for Loss* was reported. Assuming a radiation loss of 5 per cent, the average net loss due to *unaccounted-for* items would be approximately 10 per cent, which was the value assumed for the present tests.

Deducting the sum of the total losses from 100, the overall thermal efficiency of the two oil-fired heating plants was 51 and 65 per cent.

GENERAL SUMMARY OF OIL-FIRED TESTS

Relation of Equipment to Performance. The overall efficiencies shown by these two oil-fired heating plants were lower than were to be expected of these fuels, but the reasons for the low efficiencies were fairly obvious. The air-fuel ratio was, as pointed out, much higher than should be maintained. If the burners were properly adjusted when they were installed, they had been allowed to get out of adjustment with consequent loss of efficiency. Before

[10] Comparison of Oil and Gas Firing in a Heating Boiler, L. E. Seeley and E. J. Tavanlar, A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1933.

TABLE 4—PERFORMANCE DATA
OIL-FIRED PLANTS

ITEM No.		*On* PERIOD	*Off* PERIOD	*On* PERIOD	*Off* PERIOD
	GENERAL INFORMATION				
1	Case number	11		8	
2	Type of house	Wood		Brick	
3	Number of rooms	7		9	
4	Maximum design heat loss of building—Btu per hr	65,937		78,366	
5	Type of heating system	Warm-Air		Steam	
6	Leader area—sq in	608		...	
7	Radiation—sq ft	...		392	
8	Date of test	Jan. 13–17, 1936		Feb. 3–7, 1936	
9	Duration of test—hrs	112		123	
10	Degree-days in test period	148		222	
11	Average daily mean temperature—F.	33		23	
	FUEL				
12	Kind (A.P.I. Number)	No. 3 Oil		No. 1 Oil	
13	Calorific value (gross)—Btu per gal.	141,600		133,534	
14	Calorific value (gross)—Btu per lb.	19,300		19,752	
15	Total fuel fired—gas	...	46	...	89
	DRAFTS				
16	Draft in furnace—in. water	0.105	.085	.043	.044
17	Draft at smokehood—in. water	0.121	.088	.039	.038
	FLUE GASES				
18	CO_2—per cent	7.4	...	6.7	...
19	O_2—per cent	10.7	...	11.4	...
20	CO—per cent	0.0	...	0.0	...
21	N_2—per cent	81.9	...	81.9	...
22	Dry flue gas per lb of fuel—lb	28.43	...	31.34	...
23	Excess air—per cent	98	...	111.5	...
	TEMPERATURES				
24	Flue gases—F	547	...	866	...
25	House living quarters, F	69		71	
26	House living quarters—relative humidity—per cent	43		29	
	HOURLY RATES				
27	Fuel burned (average)—gal per hr	0.41		0.72	
28	Heat input (average)—Btu per hr	58,056		96,545	
	BURNER OPERATION				
29	Average length *On* period—min	22.2	...	31.2	...
30	Average length *Off* period—min	...	42.6	...	51.6
	POWER				
31	Total Power—kwhr	...	6.0		3.4
32	Power per 100 gal oil fired—kwhr	13.0		3.8	
	HEAT BALANCE—PER CENT				
33	Losses due to moisture	8.0	...	8.8	...
34	Loss due to heat in dry flue gas	16.8	...	30.3	...
35	Unaccounted losses (assumed)	10.0	...	10.0	...
36	Total losses	34.8	...	49.1	...
37	Overall efficiency (by difference)	65.2	...	50.9	...
38	Total	100.0	...	100.0	...

TABLE 5—SUMMARY OF TEST DATA

	Group 1 Bituminous Coal Hand-Fired			Group 2 Semi-Bitum. Coal Hand-Fired		Group 3 Coke Hand-Fired		Group 4 Bituminous Coal Stoker-Fired			Group 5 Gas-Fired			Group 6 Oil-Fired	
Case Number	13	26	32	14	4	1	9	2	5	10	6	33	30	11	8
Type Heating System[a]	W.A.	St.	W.A.	W.A.	W.A.	W.A.	W.A.	St.	H.W.	W.A.	W.A.	St.	W.A.	W.A.	St.
CO_2—Per Cent	7.8	7.8	4.0	6.0	5.7	6.4	10.3	5.3	9.3	6.9	3.3	8.1	8.8	7.4	6.7
Excess Air in Flue Gases—Per Cent	105.0	108.5	310.0	182.0	188.5	182.0	86.7	206.5	76.8	135.8	196.8	37.8	25.2	98.0	111.5
Temperature of Flue Gases—F	454	574	363	428	463	219	475	358	882	749	346	427	687	547	866
Heat Balance—Per Cent															
Losses Due to Moisture	4.7	4.6	5.9	4.0	4.0	0.7	0.7	4.9	5.4	5.0	10.3	10.6	11.5	8.0	8.8
Losses Due to Heat in Dry Flue Gases	16.8	27.8	27.6	21.5	24.4	9.6	17.0	21.1	32.3	34.7	15.6	8.3	13.4	16.8	30.3
Losses Due to Carbon Monoxide	4.0	1.4	1.3	0.9	1.0	3.2	2.0	0.0	1.2	0.0	...	...	...	...	...
Losses Due to Unconsumed Carbon in Ash	1.0	0.4	3.8	4.6	3.3	0.6	0.5	0.0	0.0	0.3	...	...	...	...	...
Unaccounted Losses (assumed)	20.0	20.0	20.0	15.0	15.0	10.0	5.0	5.0	5.0	5.0	5.0	5.0	5.0	10.0	10.0
Total Losses	46.5	54.2	58.6	46.0	47.7	84.1	25.2	31.0	43.9	45.0	30.9	23.9	29.9	34.8	49.1
Overall Efficiency (by difference)	53.5	45.8	41.4	54.0	52.3	75.9	74.8	69.0	56.1	55.0	69.1	76.1	70.1	65.2	50.9
Total	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0

[a] W.A.=Warm Air.
St.=Steam.
H.W.=Hot Water.

leaving each of these test houses, the burners were adjusted to give 10.0 and 9.5 per cent CO_2, which represented a gain in efficiency of 7 to 10 per cent for each unit.

The high flue-gas temperatures found in Case 8 were another example of the practice cited under the discussion of the performance of the stoker-fired tests. The burner and its rate of operation were not suited to the design of the boiler, and inefficient heat absorption resulted.

Although these results are not to be considered as representative of best practice, they are typical of the operation of many heating plants burning fuel oil. As a result of a survey [11] of oil-fired heating systems conducted in Madison, Wisconsin, it was stated:

"A survey of some 140 residence installations indicated a stack loss of 27 per cent as determined by a CO_2 average of 7.8 per cent and an average flue-gas temperature of 634 F. . . . A wide variation in stack losses was found indicating that many installations may be rated as excellent and others as unacceptably poor. The largest loss found in the survey was 46 per cent and the smallest was 10 to 11 per cent."

GENERAL SUMMARY

For purpose of convenient reference in discussion of the general results of the investigation, Table 5 presents a summary of the most significant data and the heat balances of the 15 test installations.

Hand-Fired Solid Fuel

The two principal sources of loss in firing bituminous or semi-bituminous coals by hand are the *unaccounted-for losses,* which include the losses in soot, tar, and gaseous hydrocarbons, and the dry-gas loss. The former loss is high because of the impossibility of burning the gases which are rapidly evolved when a heavy charge of fuel is fired at infrequent intervals, which is the common practice in domestic firing. The dry-gas loss is high because of the high excess of air generally carried; this, too, results from the infrequent attention to the fire.

The soot, tar, and hydrocarbon loss is so high that it offers a great field for improved devices or methods for firing boilers and furnaces by hand. The purpose of the coking, side-bank, alternate, and conical methods of firing is to reduce this loss.

The efficiencies found for coal fall within the lower limit of the range of efficiencies reported by the Bureau of Mines.[12] The average of the three values for bituminous coal is 47 per cent. It may be assumed that the efficiency of hand-fired residential installations using bituminous coal is an average value of 45 per cent. Some will undoubtedly obtain better results than this; a few may do worse.

The average of the two values for semi-bituminous coal is 53 per cent. It is higher than found for bituminous coal principally because of the justified assumption of the lower *unaccounted-for loss.* A probable average efficiency for this type of coal is 50 per cent.

[11] Loc. cit. See note 2.
[12] Loc. cit. See note 2.

With coke as a fuel, the loss in tar, soot, and hydrocarbons is practically eliminated because of the low content of volatile matter. Because of the absence of any caking action, more uniform fuel beds can be maintained and a lower excess of air is possible than when firing coal. The efficiencies actually found with coke in the two test installations were 75 per cent. It was recognized that the installations were better maintained and more carefully fired than the average. Hence an average efficiency for coke might be assumed to be about 70 per cent.

Although this assumption may seem high, it must be remembered that if coke is to give satisfaction and continue to be used, the heating equipment must be maintained in better condition, particularly as to control of drafts, and somewhat more intelligent, though not necessarily more frequent, attention must be given than can be done with coal. Hence a continued, satisfied user of coke is quite likely to be obtaining a relatively high efficiency.

Stoker-Fired Coal

The losses due to soot, tar, and gaseous hydrocarbons are largely reduced when bituminous coal is burned on the small underfeed stoker. The stokers tested did not, however, maintain an excess of air as much lower than with hand-firing as would have been expected. This indicates the necessity of proper adjustment of the air-coal ratio. The principal cause of their high loss in sensible heat in the dry flue gas was the high flue gas temperature which resulted from the application of stokers with high rates of heat liberation to boilers and furnaces designed for much lower rates. The stoker must be properly sized to the boiler or furnace, and equipment particularly designed for stoker operation is very much to be desired for new residential installations.

Excluding the one value of 69 per cent, which was principally due to a very large boiler, the average efficiency was 55 per cent, which may be assumed to be representative of this type of installation.

Natural Gas

When gas was used as a fuel, the losses to *unaccounted-for* items and the sensible heat in the dry flue gas were both low. The loss due to moisture in the flue gas was about twice that of coal because of the high hydrogen content of the gas; this is inherent to the fuel.

The efficiency found for the gas-designed warm air furnace and for the warm-air furnace with a conversion burner was 70 per cent. For the designed boiler-burner unit, the efficiency was 75 per cent. Both of these values are in the range to be expected.

Oil

The principal source of loss of heat when firing oil was in the sensible heat in the dry flue gas. This was higher than it should have been, because the excess air was high due to improper adjustment of the air-oil ratio and, as with stokers, to the liberation of heat at rates higher than those for which the fur-

nace or boiler was designed. The average efficiency found was 58 per cent; a value of 60 per cent may be assumed as typical.

EFFECT OF EQUIPMENT

The normal expectations would have been that definite trends in efficiency of warm-air furnaces, hot-water and steam boilers would have been found but, as pointed out, this was true only with gas as a fuel. The many other variables, as condition of the equipment, correctness of adjustment, and the degree of attention to the fire, obscured for the relatively few installations tested the effect of the type of heat transfer unit.

For residential heating equipment in general, therefore, the following values may be assumed as the thermal efficiencies reasonably to be expected:

	Per Cent
Bituminous coal, hand-fired	45
Semi-bituminous coal, hand-fired	50
Coke, hand-fired	70
Bituminous coal, stoker-fired	55
Natural gas, conversion burners	70
Natural gas, designed burner-boilers	75
Oil, conversion burners	60

HEATING SATISFACTION

Some of the comments of the users of the various fuels are of interest. Although the 1935-36 heating season was one of the most severe in years, all the users expressed satisfaction with the fuel they were using insofar as its ability to maintain the home comfortably heated was concerned. Neither did any of the users, even those with hand-fired installations, complain of lack of convenience of the fuel. All of those who fired bituminous coal by hand did complain of the lack of cleanliness, but only two of the six users of stokers mentioned lack of cleanliness of coal. Neither of these two was using oil-treated coal, which would have removed the cause for complaint, as others mentioned their satisfaction with the dustless feature of coal that had been treated with oil.

Thermostatic control of the rate of burning was found to be of considerable value, even on hand-fired coal furnaces, in the maintenance of uniform temperatures, especially where equipped with a limit control. The stokers were found to give excellent control of the room temperature.

COST OF HEATING

To the heating and ventilating engineer, this discussion of the sources of loss of heat and the resultant efficiencies in the combustion of various fuels for home heating is of interest from the standpoint of combustion and heat transfer. To the occupant of a home, however, the thermal efficiency is of interest only as it affects the amount of his heating bill. In addition to the data presented in this paper, data were obtained during the investigation on the costs of heating in the 15 test houses and 15 other houses. These data were

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not considered to be appropriate to the present paper, but are presented in a complete report [13] published by Bituminous Coal Research, Inc.

As, in most parts of the United States, the cost of heat in coal or coke is considerably cheaper than in gas or oil, it is evident that, despite the lower efficiencies, the cost of the heat delivered to the house will generally be lower with solid fuel than with liquid or gaseous fuels. Also it is evident that because of the higher efficiencies to be obtained with the use of coal on stokers the cost of heat on the basis of fuel costs will be less than with hand-firing. If the cost of the coal for stokers is lower than for hand-firing, as it frequently is because of the smaller size of coal used, additional savings are made. As the stoker requires power for its operation, the cost of current must be deducted, but the net saving in fuel cost is appreciable.

Properly speaking, although it is seldom considered by the home owner, the cost of heating must include the depreciation, interest, and maintenance cost on the entire heating plant. These costs are obviously lowest for hand-fired installations. Whether or not the savings due to increased efficiencies will offset the fixed charges on the automatic firing equipment will depend on the amount of fuel used which is governed by the size of the installation and the severity of the climate. For the average home in most parts of the country, coal stokers are the only form of automatic heating with which there is any saving in fuel cost as compared to hand-firing to offset the increase in fixed charges on the heating equipment. With other fuels there are not only additional fixed charges but also additional cost for fuel.

The really important advantages of automatic heating of homes are increased comfort, increased convenience, and increased cleanliness. What any given person will pay to attain some measure of these advantages will depend on his economic status and his habits of thought. It is the duty of the heating engineer to set up for the home owner a balance sheet for his local conditions which will show truly the overall costs of the available methods of heating. To do this, the probable attainable thermal efficiencies are necessary. It is the hope of the authors that the data presented in this paper will be helpful to that end.

ACKNOWLEDGMENTS

The test observations were made by Elmer White and Henry N. Ostborg of the Fuel Division of Battelle Memorial Institute. Analyses of coal and ash samples were made in the chemical laboratory of the Institute under direction of J. D. Sullivan.

Acknowledgment is made of the cooperation of the local home owners which made it possible to carry on the investigation. The assistance of Arthur Lloyd, Columbus Division, City Ice and Fuel Co., of Campbell J. Graf, Williams Oil-O-Matic Sales and Service, H. E. Irion, National Refining Co., and M. S. Roberts, Roberts Plumbing Co., in locating suitable homes burning the several fuels is also gratefully acknowledged.

The cooperation of W. T. Alexander, chief of Columbus Station, U. S. Weather Bureau, in furnishing special compilations of weather data is greatly appreciated.

[13] Technical Report No. 3, December, 1936, Efficiencies and Costs of Various Fuels in Domestic Heating, by Ralph A. Sherman and R. C. Cross, Bituminous Coal Research, Inc., Washington, D. C.

DISCUSSION

PROF. L. E. SEELEY (WRITTEN): The paper just presented is the result of much industry and of careful testing. The authors will be the first to admit the test difficulties, which are several, and the hazards of choosing an *unaccounted-for loss.* While the sources for the *unaccounted-for losses* are reliable, the question can properly arise whether or not the data are sufficient for general application. The authors assumed an *unaccounted-for* loss for oil burners of 10 per cent. Just recently, A. H. Senner reported unaccounted values for oil burners ranging from 3.0 to 6.2 per cent where the CO_2 varied between 6.9 and 9.4 per cent. Apparently the *unaccounted-for losses* will generally be low with high excess air. This example is cited merely to show the hazards involved in the method chosen by the authors and is not dictated by any bias so far as the several fuels are concerned.

It appears that the efficiency of gas and oil burners was based upon measurements during the operating period of said burners. If so, the actual efficiencies should be lower than those shown due to losses during the non-operating period. Since the rate of fuel burning for both stoker and hand-fired heaters must have varied, it would be necessary to get a weighted average of such items as flue temperature and CO_2. The data do not specifically indicate that such averages were in fact weighted averages nor does it allude to this matter in the text. The advisability of making a statement in regard thereto is suggested.

Inasmuch as these tests were run under different weather conditions it is not conclusive that the efficiencies do represent a satisfactory average for the heating plant in question and much less that they represent any average for the various fuels.

The authors' statement—"Of the several factors that govern the choice of a fuel or the furnace or boiler in which it is burned for residential heating, the efficiency is by no means the most important"—is entirely reasonable even though efficiency is the theme of their paper. The effect of listing fuels and efficiencies to be reasonably expected on page 209 seems to counteract somewhat the original sentiment. If a low and high value had been given for each fuel, it would have expressed the actual state of affairs more accurately. Such a table would be less clear-cut, especially to laymen, but the picture would be more in accord with actual facts.

It takes courage to do work of this kind in response to a demand for information when it is known that the difficulties involved are almost certain to invite criticism even from one's best friends. The writer is inclined to believe that a comparison of fuels and fuel costs should be based on many records of fuel use in the field. This would provide a study of a statistical character that would justify a certain amount of generalization.

W. E. STARK: Looking in Table 3 at the tests on the gas-fired plants, with particular reference to Case 6, which apparently is a warm-air furnace designed for use of gas; I see there a CO_2 content in the flue gases of 3.3 per cent, with a total stack loss of 30.9 per cent. That CO_2 content is absurdly low for modern appliances and indicates one or another of several things. Either the furnace was not a modern furnace, as representative of modern practice in appliances of this kind, or it was operated way under rating, which is inexcusable, or the CO_2 or flue gas analysis sample for it was taken beyond the relief opening into the draft hood.

No modern gas-designed warm-air furnace of the type that has *American Gas Association* approval today, operated at its rating, will give such absurdly low CO_2 contents in the flue gases.

L. C. PRICE:[14] The efficiencies that were shown in the last figure look a little bit

[14] Research Engr., University of Arkansas, Fayette, Ark.

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high for all season efficiencies. Were those intended to be all season averages or were they directed to the same load on the heating plants and what loads were they?

R. J. BENDER:[15] One of the gentlemen in the audience talked about the results of gas-fired installations. I would like to say something about oil-fired installations. I am sorry that my good friend, Mr. Sherman, is not here to discuss the matter a little more fully. I have had discussions with him on this subject before. It seems to me that the results given in the paper might have been representative about four years ago.

It is true indeed that about four years ago, the average oil burner installation was poorly adjusted; but since that time much progress has been made; so much interest has been shown, not only by the manufacturers and the people who install or service oil-burning installations, but also by the oil companies, that I doubt very much if the two efficiency figures indicated for the warm-air furnace and the steam boiler can be considered representative now, or even a year ago.

The very high temperature for the steam boiler shows evidence that the burner was entirely too large, or the nozzle was entirely too large for the boiler, an error which was indulged in quite frequently years ago, but which no manufacturer would dare commit today, with the very hard competition he has to contend with on the part of other fuel-burning equipment manufacturers.

Another thing I would like to bring out is that these tests were conducted in Columbus, O., where the market for oil burners is not highly competitive: some installations may be put in and if they work fairly satisfactorily, and, as said in the latter part of the paper, if they are clean and comfortable to the owner, they may pass. In a city such as Chicago, Detroit, or some other towns where the oil burner market is highly competitive, things like this would not happen. As soon as installations such as these became known to the trade, there would be oil burner men after them to change burners, or oil companies offering to have the burners adjusted properly.

I am afraid that if this paper, after receiving the seal of approval of the Institute, or the sanction of this Society, is published, some manufacturers may get hold of it and do a lot of damage.

Another thing Mr. Cross said in his comment, that has been left out of the paper, is that boiler-burner-units and furnace-burner-units being very efficient, nowadays constitute the greater part of the sale. All of you who have looked through the January issue of *Fuel Oil Journal* have seen the tremendous progress that has been made in the sale of oil-burner-units: they have efficiencies above those indicated in the paper.

To summarize, we always take today in the oil industry an efficiency of 70 per cent as being a fair representative average for oil burning plants, as compared to coal, stoker-fired and hand-fired, or gas. We take no exception to the efficiencies which have been assigned to these other fuels in the paper.

H. P. REID: I wish to compliment Messrs. Sherman and Cross on this work; however, I feel it is regrettable that so many of the burners were so poorly adjusted. As I understand, the tests were made on the burners and on the installations as found, and after the tests were made, service or advice was given to the home owners for proper adjustment of the equipment. It seems to me that to really make this a true picture of conditions and possibilities, another test should have been made after the equipment had been adjusted, otherwise, the information given is certainly misleading in a great many ways.

I wish to call your attention to a few figures. Those of you who are familiar

[15] Engr., Sinclair Refining Co., Chicago, Ill.

with Órsats and Órsat analysis, might not be interested, but a great many have not studied many Órsat analyses. In round figures when natural gas, or gas of this general composition, is burned you expect about 11.5 or close to 12 per cent of CO_2 from Órsats with perfect combustion—neither excess air nor deficiency of air; with oil about 15.8 per cent and with coal about 18.5 per cent. As against these, the natural gas burners reported in this paper produced as low as 3.3 per cent of CO_2—very poor adjustment. Not only does that poor adjustment lend itself to poor efficiency, but it also causes low flame temperatures.

While I have not made any effort to check the calculations, it looks peculiar to find 70 per cent boiler efficiency with natural gas fired with extremely low combustion efficiencies, as shown by gas analyses, while very low boiler efficiencies were obtained with relatively good combustion efficiencies with other fuels. If good efficiencies could be obtained with natural gas, certainly they should with other fuels also.

Another point not brought out is the effect of the radiant energy from the burning bed of solid fuels, and the lack of this radiant energy from oils and gas. This, by the way, has considerable effect upon the capacity of the furnace. Leaving the heating furnace for a moment, and going to industrial furnaces—in a great many cases, we lose 10 per cent overall capacity when we change from pulverized coal to natural gas firing. Efficiencies when using oil lie between those for coal and natural gas.

Another point that I think needs careful consideration by the technical men is the real effect of moisture in furnace gases. In the heat balance as shown here, which is a typical heat balance, in general approved by the *American Society of Mechanical Engineers* and other scientific bodies, all moisture in the gases is treated as a loss. A great many of you people know that the old fireman always wetted down his coal and he got better steaming when so doing. Why? Let me give you an analogy.

In the cement industry where we are burning pulverized coal, natural gas and oil in our kilns, we took a coal burner and injected live steam with the fuel. After trying different percentages we found that 3½ per cent, by weight, of steam to fuel, resulted in a fuel saving with the same production and same heat treatment in the furnaces of more than 6 per cent. The gases from this furnace passed through waste heat boilers and with the 6 per cent reduction in fuel consumption, steaming in the waste heat boiler increased appreciably.

Only the CO_2 and H_2O of all gases are radiant at elevated temperatures as in furnaces.

As the percentages of either of these increase in the gaseous products of combustion the radiant effect of those gases is increased. Therefore, if these gases are diluted with air, not only is the temperature in the furnace reduced but the radiant capacity of the flame is also reduced.

I submit that while this paper is extremely valuable in showing conditions as found, one of its main values is to point out the poor adjustment and poor use of fuels in residence heating.

R. C. CROSS: I will try to be as brief as possible. The matter of Case 6, showing 3.3 per cent CO_2 is a natural question to bring up. This was a Bunsen type burner with a space around the burners, having no controllable shutters, and the CO_2 was very low. For the benefit of Mr. Stark, I will say that I feel quite sure the gas sample was taken correctly. This was a modern furnace approved by the *A.G.A.*

I will go to Mr. Reid's comments, in connection with the same installation. The fact that we got such a high efficiency is largely governed by the fact that the flue gas temperature and sensible heat loss of the flue gases was low.

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Referring to Mr. Price's discussion, the efficiencies which are given should be taken and used with some consideration of conditions. We don't set them up as entirely representative but they are typical. You might get entirely different results from 15 other houses. We merely selected these houses at random. The efficiencies are efficiencies for the period of test, and they are not seasonal. Some were run in mild weather and some in extremely cold weather.

I was interested in Mr. Bender's comments. What he says is true. The development of the art of oil burners has come along rapidly. If you will refer to the paper, you will note the conditions are for conversion installations, and I believe 65 per cent would be a top figure for the average conversion installation. I recall having had a record of a large number of field installations, some 350, I believe, where short tests were run, and in that number only 6 per cent could be considered acceptable from all angles. I think the efficiency would be nearer 60 per cent than 70 for the average oil conversion installation, although in some of the very fine units available you may get up to 80 per cent efficiency.

In regard to the question of boiler-burner units, we made an effort to find one, but in Columbus we found none which we could use.

Mr. Reid discusses some theoretical CO_2 values of perfect combustion. That is well and good but, on the other hand, I would be hesitant to inject it into a meeting of this sort. It is possible in the laboratory, but it might be misinterpreted if a gas-fired installation were raised from 3 per cent to 12 or 13 per cent, when the field averages might run between 8 and 10 per cent.

I don't think the effect of introducing moisture into the furnace is anything we can consider at this time.

No. 1057

PERFORMANCE OF OIL-FIRED, WARM-AIR FURNACES IN THE RESEARCH RESIDENCE

By A. P. KRATZ * AND S. KONZO ** (*MEMBERS*), URBANA, ILL.

THE investigations in forced-air heating in the Research Residence at the University of Illinois during the period from 1932 to 1934 were largely confined to studies of the characteristics of the forced-air heating system,[1] to studies of automatic controls,[2] and to studies of the effectiveness of storm windows [3] under actual service conditions. For these investigations the fuel used was anthracite. The investigation for the heating season of 1934-1935 was undertaken to determine the performance and operating characteristics of a warm-air furnace equipped with a conversion oil-burning unit. For the heating season of 1935-1936 the conversion oil-burning furnace unit was replaced with a warm-air furnace designed specifically for oil combustion, and the performance and characteristics of the furnace were compared under identical conditions of operation with those obtained with the conversion unit.

DESCRIPTION OF THE RESEARCH RESIDENCE AND HEATING EQUIPMENT

The Research Residence, shown in Fig. 1, together with the forced-air heating system has been described in previous papers.[2, 3] The residence is a three-story structure of standard frame construction. The wall section consists of weather-boarding, building paper, sheathing, 6-in. studding, wood lath, and plaster with rough sand finish. The walls are not insulated and no weather stripping is used at the windows and doors. All the tests reported in this paper were made with ordinary, double-hung windows tightly locked.

The total space heated during these tests consisted of three rooms, a sun room, a breakfast nook, and a hallway on the first story; three rooms, a bathroom, and a hallway on the second story; and two rooms, a bathroom, and a hallway on the third story. The total volume of this heated space, from which

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** Special Research Associate, Engineering Experiment Station, University of Illinois.

[1] University of Illinois, Engineering Experiment Station, Bulletin 266 by A. P. Kratz and S. Konzo, 1934.

[2] Automatic Controls for Forced-Air Heating Systems, by S. Konzo and A. F. Hubbard, A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 37.

[3] Fuel Saving Resulting from the Use of Storm Windows and Doors, by A. P. Kratz and S. Konzo, A.S.H.V.E., TRANSACTIONS, Vol. 42, 1936, p. 87.

Presented at the 43rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, St. Louis, Mo., January, 1937, by S. Konzo.

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the basement was excluded, was approximately 17,540 cu ft. The calculated heat loss was approximately 137,500 Btu per hour at an indoor-outdoor temperature difference of 70 F, and approximately 159,000 Btu per hour at an indoor-outdoor temperature difference of 80 F. The residence is completely furnished, and during the heating season it was occupied by four people.

The heating plant consisted of a warm-air furnace used in connection with a forced-air heating system. The duct system is shown in Fig. 2. Three cold-air returns were used which were connected to a cold-air box above the inlet to a centrifugal type fan. The furnace was placed at the east end of the basement, and the warm-air registers were served from two main trunk systems. The furnace used during the heating season of 1934-1935, shown in Fig. 3, was

FIG. 1. VIEW OF RESEARCH RESIDENCE IN URBANA, ILL.

of the cast-iron, circular-radiator type, having a 27-in. firepot and 23-in. grate, and was equipped with a casing 50-in. in diameter. In order to accommodate the gun-type oil burner which was inserted through the ash-pit door of the furnace, the grates were removed and firebrick were arranged to form a cylindrical combustion space as shown in Fig. 3. Hereafter this combination of the furnace and burner will be referred to as the conversion unit.

During the heating season of 1935-1936 the furnace used, shown in Fig. 4, was of steel construction especially designed for oil burning. This furnace will be referred to hereafter as the oil-burning furnace. The burner was the same gun-type oil burner which was used in the conversion unit. Hence, any differences in performance of the two furnace installations could be attributed to the differences in the furnace and not to the differences in the type of oil burner used. The fan and furnace were built into a compact unit which was housed in an insulated casing. The values for the heating surfaces of the two furnaces are listed in Table 1, from which it may be noted that the oil-burning

furnace had approximately 45 per cent greater heating surface than did the conversion unit.

The fuel oil burned was a No. 3 grade furnace oil having a specific gravity of approximately 36 deg Baumé and a heating value of approximately 19,750 Btu per lb.

The control [4] of the heating plant was accomplished by means of a room thermostat operating to start and stop the oil burner and the circulating fan. This room thermostat was used in conjunction with two bonnet thermostats, one of the latter operated to stop the burner when the bonnet temperature exceeded a predetermined value, and the other operated to stop the fan when the bonnet temperature became less than a predetermined value. The room thermostat was located on an inside wall of the dining room at a height of 60 in.

TABLE 1—AREAS AND DIMENSIONS OF FURNACES

HEATING SURFACE SQ FT	CONVERSION UNIT	OIL-BURNING FURNACE
Ashpit	11.38	....
Firepot	8.10	....
Tubes	....	59.5
Combustion Section	18.87	23.6
Radiator	31.93	19.0
Total, sq ft	70.28	102.1
Grate Area, sq ft	2.88	....
Refractory top section	Diameter	
inside dimension, in.	17.0	14.5 x 26.5
Combustion space,[a] cu ft	2.36	2.56

[a] The combustion space is regarded as that space enclosed within the refractory bricks.

from the floor, and was adjusted to maintain an air temperature of approximately 71 F at this level in all of the rooms of the Residence.

METHOD OF CONDUCTING TESTS

Tests Under Continuous Operation

In order to determine the capacity at the bonnet and the thermal efficiency of the furnaces for various conditions of operation, tests were made under conditions of thermal equilibrium with both the furnace and the fan operating continuously. In these tests, the temperature of the inlet air was maintained at approximately 70 F, and the quantity of circulating air, the rate of oil input, the draft in the combustion chamber, and the amount of air required for combustion were maintained at predetermined values during the two-hour test period. These tests, which corresponded to the usual laboratory tests for the determination of the heat output of a heating unit, provided data for the heating unit only, without reference to the structure in which it was placed.

[4] See Control type IV described in reference item 2.

Leader Ducts					Stacks or Throats			Registers	
No.	Name	Size	c.f.m.	vel.	Floor	Size	vel.	Size	vel.
(1)	N.W. Bedrm.	8x8½	239	506	2nd	5½x13	481	10x12	412
(2)	Kitchen	7x8	180	463	1st	7x14	265	12x14	215
(3)	N. Trunk, 1	8x12¾	419	592					
(4)	Sun Rm. N	5½x13	241	485	1st	5½x13	485	10x12	416
(5)	Bath	3x10	56	269	2nd	3x10	269	8x10	152
(6)	Sun & Bath	8x10	297	535					
(7)	S.W. Bedrm.	8½x5½	167	478	2nd	3½x12	572	9x12	325
(8)	N. Trunk, 2	8½x12	464	636					
(9)	E. Bedrm.	6x12	348	696	2nd	6x12	696	12x14	600
(10)	Main, N. Tr.	8x32	1231	692					
(11)	Main, S. Tr.	8x32	1330	748					
(12)	E. Dorm	3x10	148	710	3rd	3x10	710	8x10	402
(13)	Liv. Rm.	5½x13	337	679	1st	5½x13	679	12x14	582

Leader Ducts (con.)					Stacks or Throats			Registers	
No.	Name	Size	c.f.m.	vel.	Floor	Size	vel.	Size	vel.
(14)	Hall	7½x14	305	418	1st	7½x14	418	12x14	365
(15)	S. Trunk	8½x14	540	654					
(16)	W. Dorm.	3x10	152	730	3rd	3x10	730	8x10	413
(17)	Sun & Din.	7x14	388	570					
(18)	Din. Rm.	5½x13	147	296	1st	5½x13	296	10x12	254
(19)	Sun Rm. S.	5½x12	241	503	1st	5½x12	503	10x12	416

●-Represents Location of Squeeze Damper

Return Ducts					Grilles*	
No.	Name	Size	c.f.m.	vel.	Size	vel.
(20)	Liv. Rm.	8X14	570	713	8X30	611
(21)	Hall-1	9X12½	590	755	6X40	632
(22)	Hall-2	9X13½	590	686	6X40	632
(23)	Sun Rm.	9X14	570	652	8X30	611

*Wood Grilles (56% Free Area)

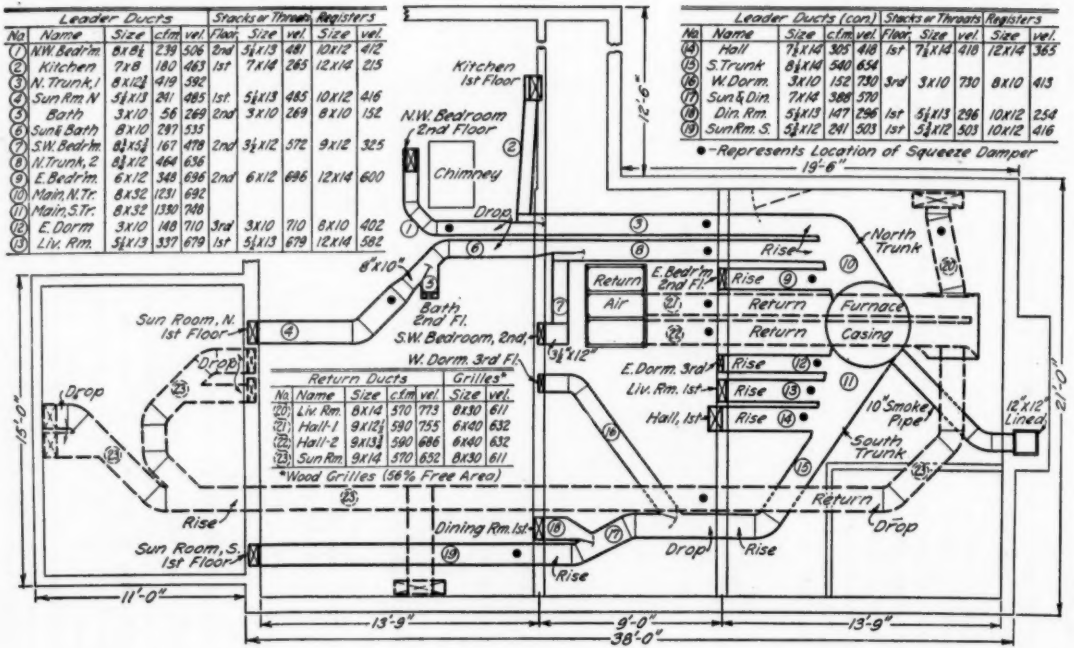

FIG. 2. PLAN OF DUCT SYSTEM IN BASEMENT

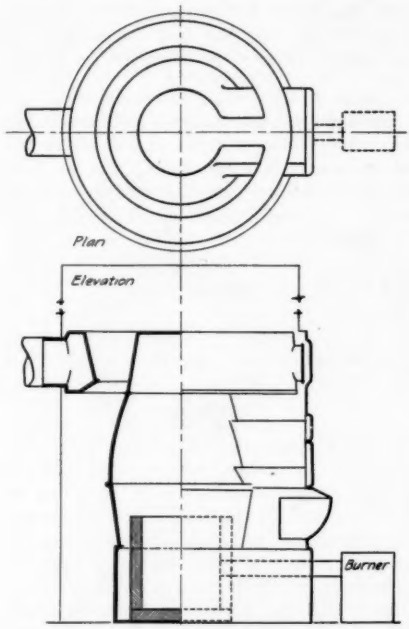

FIG. 3. DIAGRAM OF CONVERSION OIL-BURNING UNIT

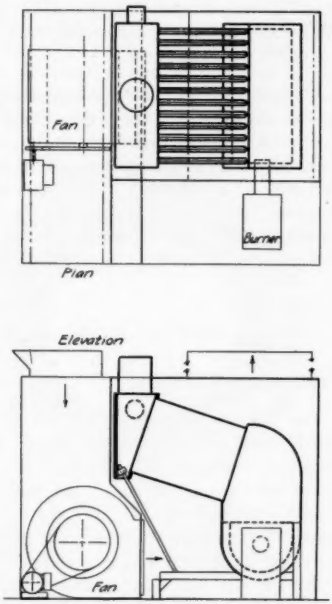

FIG. 4. DIAGRAM OF OIL-BURNING FURNACE

The quantities referred to as *capacity, combustion rate, bonnet efficiency,* and *overall house efficiency* are defined by the given equations:

Capacity, Btu per hour put into air $= W \times 0.24 \times (T_b - T_1)$

Heat liberated in furnace, Btu per hour $= C \times H$

Bonnet efficiency, per cent $= \dfrac{W \times 0.24\ (T_b - T_1) \times 100}{C \times H}$

Overall house efficiency, per cent $= \dfrac{[(C \times H) - \text{Flue loss up chimney}]\ 100}{C \times H}$

in which:

W = weight of air flowing in system per hour, lb.
0.24 = mean specific heat of air.
C = combustion rate, pounds of oil burned per hour.
H = calorific value of oil, Btu per pound.
T_1 = temperature of air at inlet to furnace, F.
T_b = temperature of air at furnace bonnet, corrected for radiation, F.

Tests Under Service Conditions

The tests made under service conditions differed from the tests made under continuous operation in that the operation of the burner and fan was automatically controlled to maintain a temperature of 71 F in all the rooms of the

house rather than to maintain a constant capacity at the furnace bonnet. Therefore, in these tests the operation of the burner and fan was intermittent in character and not continuous, and the test periods were 24 hours in length instead of 2 hours.

Observations of fuel consumption, flue gas analyses, power input to the fan motor and burner motor, flue gas temperatures, temperatures of the air in the ducts and at the registers, and room temperatures were made daily at 7:00 a.m., 11:00 a.m., 4:00 p.m., and 10:00 p.m. The total times of operation of the fan and of the burner for a 24-hour period were obtained by means of self-starting, electric clocks placed in the electrical circuits of the motors. In addition, continuous records of temperatures and CO_2, such as shown in the typical records in Fig. 5, were obtained for each 24-hour period.

RESULTS OF TESTS UNDER CONTINUOUS OPERATION

Preliminary Statement

No attempt was made to determine virtual capacities or efficiencies of the plant with intermittent operation of the fan. Average temperatures and air weights would be meaningless unless the various readings were weighted in some way proportional to the length of *on* and *off* periods. Periods during which temperatures were increasing or decreasing while air weights remained constant complicated the situation, and make it impossible to determine any proportionality factor based on time. Furthermore, all quantities were not involved to the first power alone, and the average of a number of instantaneous results involving multiplication, division, addition, and subtraction indiscriminately is not numerically equal to the result obtained by the same form of calculation applied to the averages of the several factors, particularly when these factors are varying from zero to a maximum. Hence the actual efficiency and capacity of the furnace on intermittent fan operation are indeterminate, and a statement of such quantities based on arbitrary approximations would be meaningless. The performance of the plant was therefore studied under conditions of continuous fan operation.

In the case of a gravity warm-air furnace, in which a unique relationship exists between the combustion rate, the air volume, and the register tempera-

TABLE 2. FACTORS AFFECTING CAPACITY AND EFFICIENCY OF AN OIL-BURNING, WARM-AIR FURNACE

I. Combustion characteristics
1. Type of burner
2. Rate of oil input
3. Percentage of CO_2 in flue gas
4. Conditions of draft in combustion chamber
II. Furnace characteristics
5. Amount and location of heating surfaces
6. Size and shape of combustion space
7. Quantity of air circulated

ture, the capacity will be a definite value for any given furnace installation if the combustion rate alone is specified. However, in the case of a forced-air furnace system, no such definite relationship exists between combustion rate and air volume, and as a consequence the capacity is dependent on more than the combustion rate alone. These studies of performance characteristics under conditions of continuous operation were, therefore, undertaken to obtain the

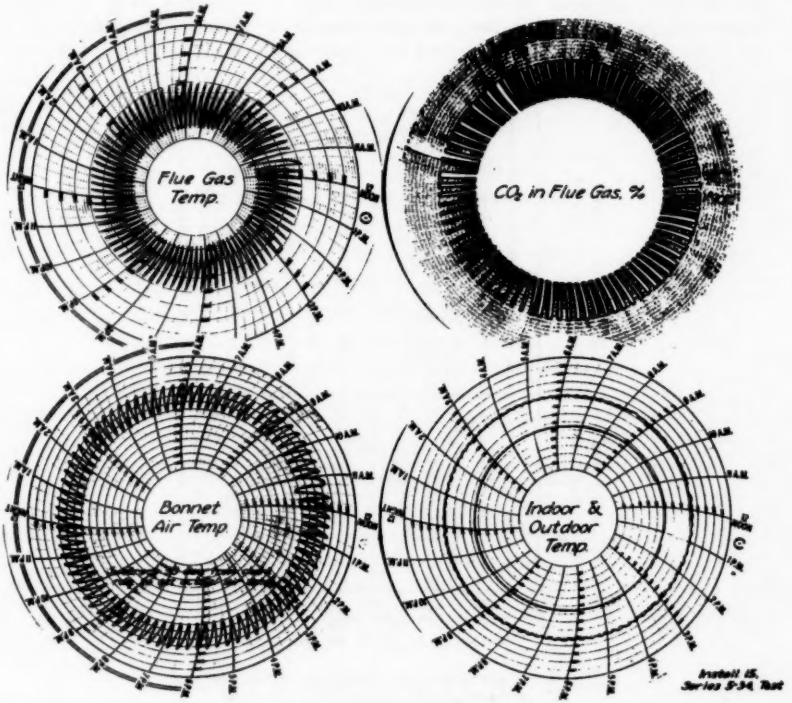

FIG. 5. TYPICAL RECORDS SHOWING OPERATION WITH CONVERSION OIL FURNACE USED WITH FORCED-AIR HEATING SYSTEM IN RESEARCH RESIDENCE.

relationships existing among some of the factors that determine the capacity of oil-burning furnaces. In general, the capacity and efficiency of oil-burning furnaces are dependent upon seven factors, which may be grouped into two classifications as shown in Table 2.

It may be noted from Table 2 that for any given installation of burner and furnace, items 1, 5, and 6 are fixed, and items 2, 3, 4, and 7 are independently variable. However, in all of the tests reported in this paper, the draft in the combustion chamber (item 4) was maintained by means of an automatic draft regulating damper at a value of 0.02 in. of water, which was sufficient to insure

the positive removal of the flue gases. Hence, the studies on furnace performance consisted of tests in which the rate of oil input, the percentage of CO_2 in the flue gas, and the quantity of air circulated were independently varied in magnitude.

Variations in Air Quantity

The results obtained from the conversion unit are shown in Fig. 6. For these tests, the CO_2 in the flue gas was maintained constant at a value of 9.5 per cent, and the effect of varying the quantity of air circulated over the

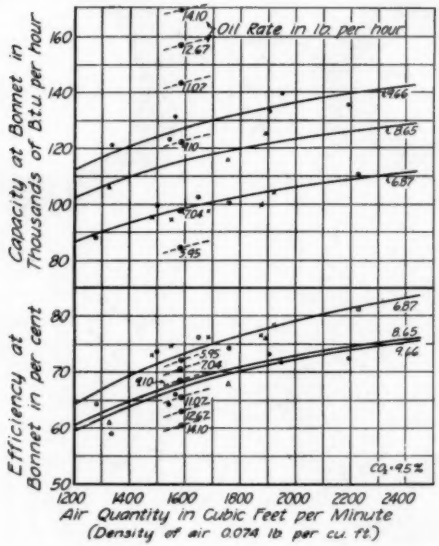

FIG. 6. PERFORMANCE CURVES OF CONVERSION UNIT IN FORCED-AIR HEATING SYSTEM

heating surfaces of the furnace was studied. The results shown are typical and are similar in nature to those obtained from tests on the oil-burning furnace installation made under similar conditions of operation. The full lines in Fig. 6 represent results obtained with constant rates of oil input to the furnace. It may be noted that for a given rate of oil input, both the capacity and efficiency of the furnace increased as the quantity of air circulated over the heating surfaces was increased. For example, at a 9.66-lb oil input rate, the capacity was increased from 120,000 Btu per hour to 140,000 Btu per hour when the air quantity was increased from approximately 1380 cfm to 2240 cfm. This represented an increase in capacity of approximately 17 per cent for an increase in air quantity of approximately 62 per cent. The corresponding in-

crease in bonnet efficiency under these conditions was from 63 per cent to 74 per cent.

It may also be noted from Fig. 6 that the increase in capacity for a given increase in air quantity became smaller as the air quantity was increased. For example, an increase in air quantity from 1400 cfm to 1600 cfm resulted in an increase in capacity of 6500 Btu per hour; whereas an increase in air quantity from 2000 cfm to 2200 cfm resulted in an increase in capacity of only 2500 Btu per hour. It is apparent from these tests that in order to facili-

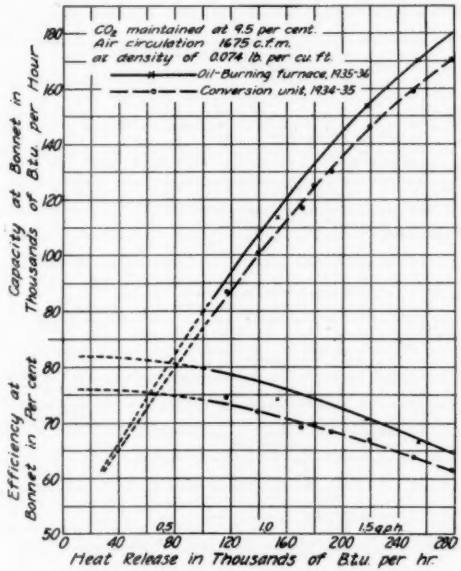

FIG. 7. PERFORMANCE CURVES OF OIL FURNACES IN RESEARCH RESIDENCE

tate the transfer of heat from the heated surfaces to the circulating air, an adequate circulation of air over the heating surfaces should be provided.

Variations in Oil Input Rate

The performance curves in Fig. 6 show also that for a constant quantity of air circulated, the capacity and efficiency of the furnace are dependent upon the rate of oil input to the burner. In order to show more clearly the relationship between oil input and capacity, the values for an air quantity of 1675 cfm shown in Fig. 6 have been plotted in Fig. 7. Similar curves for the oil-burning unit are also given. The broken lines represent results obtained

with the conversion unit, and the full lines represent results obtained with the oil-burning furnace.

It may be noted for both furnace units that the capacity at the bonnet increased as the rate of oil input, or its equivalent which is the rate of heat release in the furnace, was increased. In the case of the conversion unit, an increase in the heat release of from 120,000 Btu per hour to 240,000 Btu per

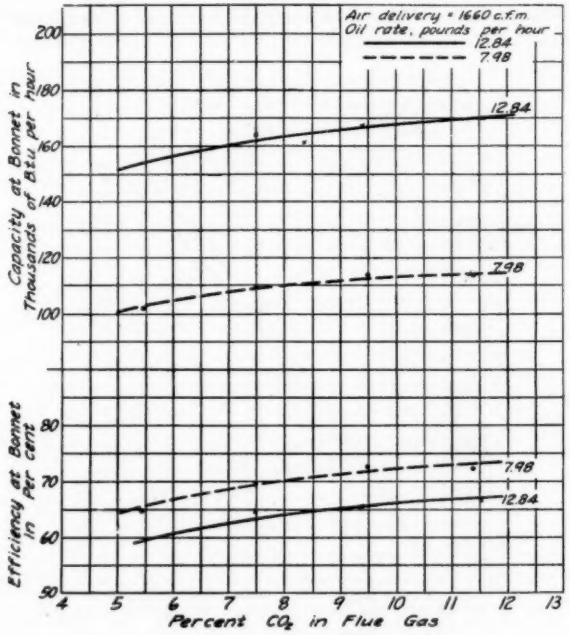

FIG. 8. PERFORMANCE OF AN OIL-BURNING FURNACE WITH VARIOUS PERCENTAGES OF CO_2 IN THE FLUE GAS

Research Residence installation 16, forced-air system. Furnace designed for oil combustion using gun burner. Tests conducted with continuous operation of burner and fan. Season 1935-36.

hour, or a difference of 120,000 Btu per hour, was accompanied by an increase in capacity of from 87,500 Btu per hour to 155,500 Btu per hour, or a difference of 68,000 Btu per hour.

In any given furnace installation, in which a constant value of CO_2 was being maintained, an increase in the amount of heat release in the furnace resulted in higher temperatures of the flue gas and hence in larger flue gas losses per pound of fuel. This condition is reflected in the efficiency curves of Fig. 7, from which it may be observed that the furnace efficiency decreased as the rate of heat release was increased. The tests made under actual service conditions, which are discussed in a later section, also tended to confirm the

conclusion that the most economical utilization of fuel in a given installation would be obtained with small rates of heat release in the furnace.

A comparison of the results from the conversion unit and oil-burning furnace, as shown in Fig. 7, indicated that, for a given rate of heat release in the furnace, the greater amount of heating surface of the oil-burning unit resulted in a greater amount of heat being transmitted from the furnace gases and absorbed by the air circulated. This conclusion was also confirmed by the tests made under actual service conditions.

The efficiency curves in Fig. 7 also show that for a furnace having a given amount of heating surface the best rate of oil input as defined by the heat release occurs at a value too low to be used as a practical basis for the design of oil-burning furnaces.

Variation in CO_2 in Flue Gas

The performance curves in Fig. 8 indicate that for a constant quantity of air circulated and a fixed rate of oil input to the furnace, the capacity and efficiency of the heating unit were also affected by the combustion characteristics, as indicated by the percentage of CO_2 in the flue gas. Adjustments in the fuel-air ratio were made by means of the adjustable opening in the air-inlet to the burner, and variations in the CO_2 content in the flue gas from 5½ per cent to 11½ per cent could be obtained. In no case was there any indication of incomplete combustion of the fuel.

The increase in capacity and efficiency brought about by an increase in the CO_2 content in the flue gas was not marked. For example, the efficiency curves for the 12.84-lb oil input rate indicate that the efficiency was increased from 61 per cent to 67.5 per cent when the percentage of CO_2 in the flue gas was changed from 6 per cent to 12 per cent. In other words, for each increase in the CO_2 content of one per cent, there was obtained an increase in bonnet efficiency of approximately one per cent.

However, it should be recognized that these results were obtained with a furnace in which the temperature of the flue gas at the outlet of the radiator was quite low and did not exceed approximately 500 F. In the case of furnaces which are not provided with sufficient effective heating surface, and which are fired at such a high rate of oil input that the temperature of the flue gas at the outlet exceeds approximately 500 F, any increase in the CO_2 content of the flue gas will be accompanied by much larger increases in furnace efficiency.

The stack-loss curves shown in Fig. 9,[5] serve to explain these conditions. For instance, for low values of the flue gas temperature, the relative change in the stack loss will be small even for fairly large changes in the CO_2 content of the flue gas. In other words, if oil-burning furnaces are provided with sufficient effective heating surface for a given rate of oil input, so that the leaving flue gas temperature will be low, then the unit may be expected to perform satisfactorily under diverse conditions of installation in the field, even with fairly wide deviations in the adjustment of the fuel-air ratio of the oil burner.

These tests which were made under conditions of continuous operation have

[5] Curve reproduced from U. S. Dept. of Agriculture Technical Bulletin No. 109 entitled, A Study of the Oil Burner as Applied to Domestic Heating, by Arthur H. Senner, July, 1929.

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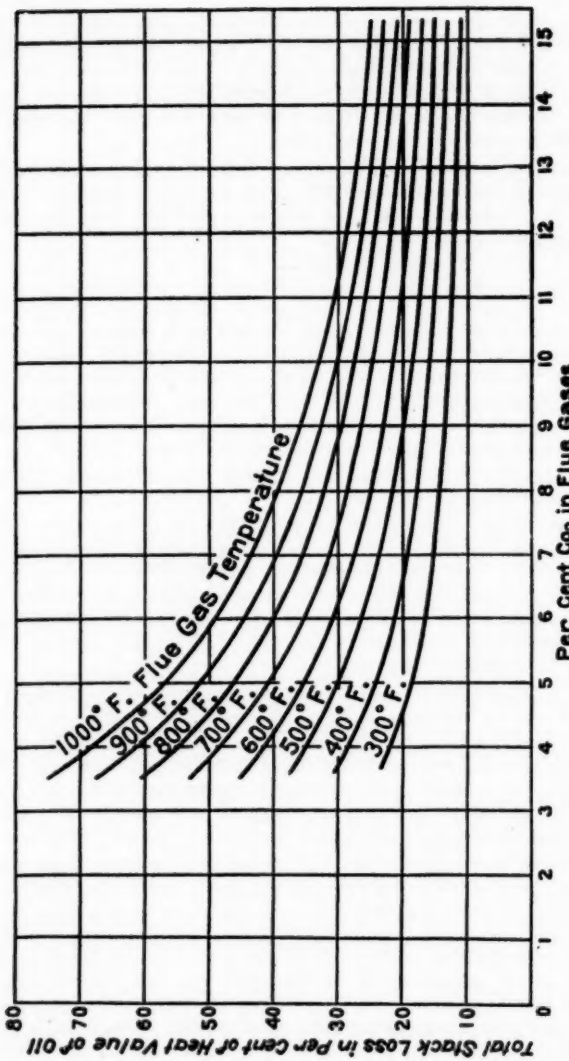


FIG. 9. STACK-LOSS CURVES FOR COMBUSTION OF DOMESTIC OIL-BURNER DISTILLATES

indicated that considerable differences in the capacity of oil-burning furnaces may be expected with variations in operating conditions. However, since capacity alone is not an index of performance under intermittent service, and since the tests were based on the maintenance of equilibrium conditions, which are seldom met with in actual service conditions, these tests were supplemented with tests made under conditions of actual service.

RESULTS OF TESTS UNDER ACTUAL SERVICE CONDITIONS

Preliminary Statement

It is evident from the preceding discussion that the results obtained from oil-burning furnaces which are operated under actual service conditions should be dependent to a great extent upon the values which are maintained for the CO_2 content in the flue gas, for the oil input rate, and for the quantity of air circulated. In the case of the air quantity, in order to maintain comparable conditions with previous tests, the speed of the fan was adjusted so that approximately 1675 cfm delivery was obtained during the on-period of the fan. During the off-period of the fan a slight *gravity* action took place and a greatly decreased air quantity was circulated. With the exception of the series of tests for which the CO_2 content in the flue gas was maintained at 11.5 per cent, all of the tests were made with the CO_2 maintained at 9.5 per cent. For one series of tests the rate of oil input to the furnace was adjusted from a minimum value of 7.3-lb per hour to a maximum value of 13.0-lb per hour. However, for those tests which were made to study the comparative performance of the conversion unit and the oil-burning furnace the rate of oil input was adjusted to 13.0-lb per hour.

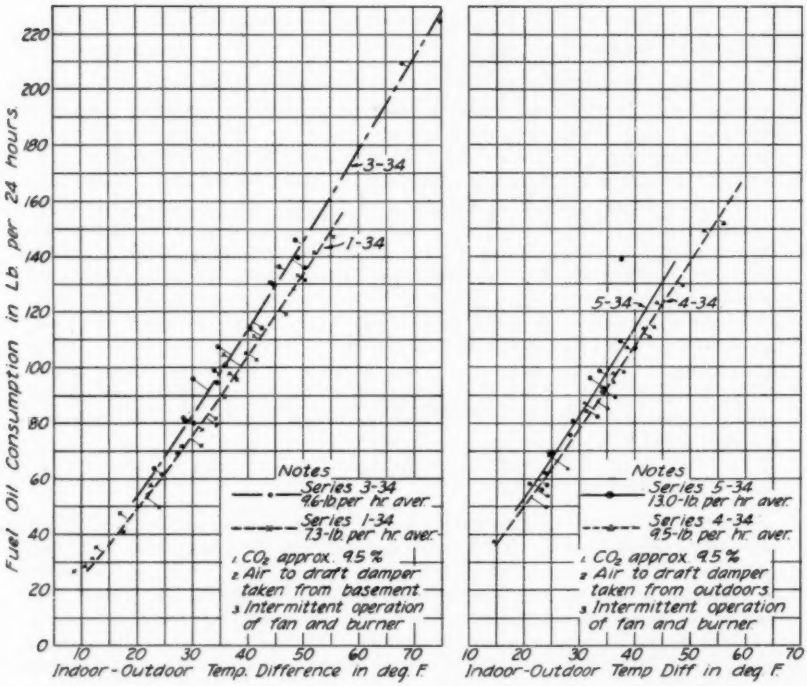
Variation in Rate of Oil Input

An extensive series of tests was conducted with a wide range of oil input rates that ranged from a minimum of 7.3-lb per hour to a maximum of 13.0-lb per hour. The heat supplied by the furnace when the burner was adjusted to maintain a 7.3-lb oil rate was not quite sufficient to offset the heat loss from the house during zero-degree weather, but was sufficient to supply the heat required during average weather conditions. The heat supplied by the furnace when the 13.0-lb rate was maintained was adequate to heat the house during the most severe weather conditions.

The weight of fuel oil required for a 24-hour period to maintain the temperature of the house at 71 F was plotted against the difference in temperature between indoors and outdoors, as shown in Figs. 10a and 10b. The curves presented in Fig. 10a are for two rates of oil input, 7.3-lb and 9.6-lb per hour; while the curves presented in Fig. 10b are for rates of 9.5-lb and 13.0-lb per hour.

Although some of the points representing the daily test results deviate somewhat from the average curve drawn through the points, because of the influences of wind and sun which cannot be shown on a curve in which the abscissa is the temperature difference alone, the curves represent the general trend of the points with a reasonable degree of accuracy. The curves show that when a constant value of CO_2 in the flue gas was maintained, the fuel consumption

increased as the rate of oil input was increased. For instance, it may be noted from Fig. 10b that the fuel requirements for a day in which the temperature difference between indoors and outdoors was 33 F were 88-lb and 94-lb for oil input rates of 9.5-lb per hour and 13.0-lb per hour, respectively. The increase in this case was of the order of 6 per cent. This indicates that the operation of the burner for relatively short periods, during which a high rate of combustion was maintained, was not as conducive towards fuel economy

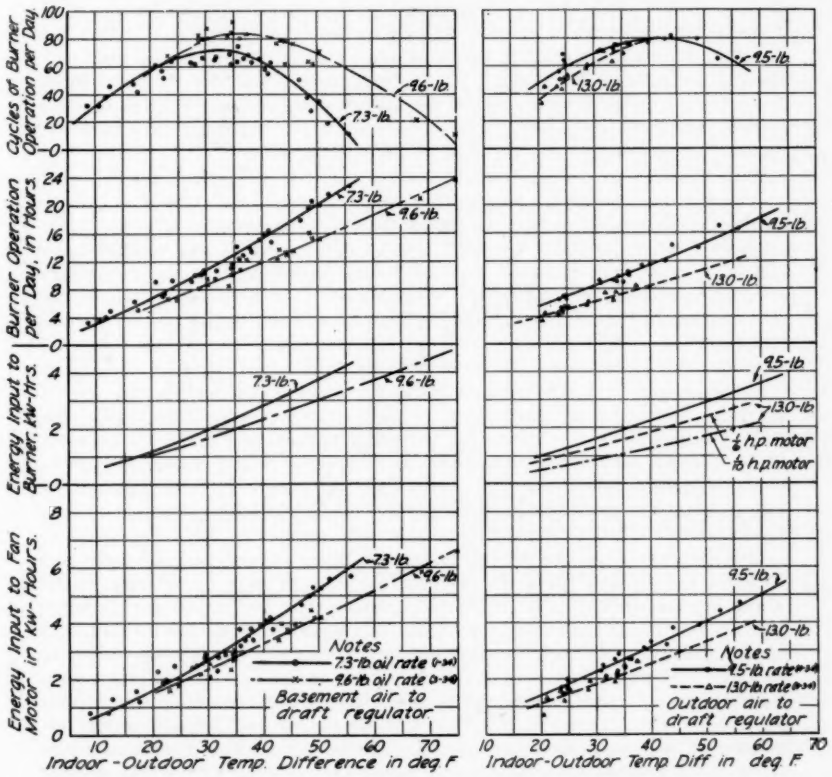


FIGS. 10a (*Left*) AND 10b (*Right*)—DAILY FUEL CONSUMPTION CURVES FOR VARIOUS RATES OF OIL INPUT INTO CONVERSION OIL FURNACE WITH FORCED-AIR HEATING SYSTEM. SEASON 1934-35, INST. 15

as the operation of the burner for longer periods, during which a lower rate of combustion was maintained. In the former case the flue losses during the periods of burner operation, and also the losses due to the heat carried up the chimney during the *off* periods of the burner, were greater than in the latter case on account of the higher temperatures of the flue gas. These statements would have to be modified, however, if there are any cases in which conditions might arise whereby the combustion process can be maintained better with high rates of oil input than with low rates.

The changes in the rate of oil input to the furnace were also reflected in the

operation of the fan and burner units in the forced-air heating system, as shown by the data in Figs. 11a and 11b. The top set of curves in Figs. 11a and 11b shows, for the same series of tests whose results were shown in Figs. 10a and 10b, the frequency of burner operation as represented by the number of cycles per 24 hours. For a given rate of oil input it may be noted that in mild



FIGS. 11a (*Left*) AND 11b (*Right*)—PERFORMANCE DATA FOR BURNER AND FAN OPERATION IN RESEARCH RESIDENCE. CONVERSION OIL FURNACE WITH FORCED-AIR HEATING SYSTEM. SEASON 1934-35. INST. 15

weather the frequency of operation of the burner increased as the temperature difference between indoors and outdoors increased. For larger values of temperature differences the burner continued in operation for longer periods and the frequency of the operating periods decreased. Finally, when the outdoor weather conditions were such that the heat developed in the furnace was just sufficient to offset the heat loss from the house, the burner operated continuously and the number of cycles per 24 hours became unity.

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It may be noted from the second set of curves in Figs. 11a and 11b that the increase in the total time of operation of the burner was practically proportional to the increase in temperature difference between indoors and outdoors. Also, for a given outdoor temperature condition an increase in the rate of oil input was accompanied by a decrease in the total time of burner operation. Furthermore, the decrease in the total time of burner operation was in turn accompanied by a decrease in the electrical energy input to the burner motor, as shown by the third set of curves. It may be observed from Fig. 11b that the substitution of a more efficient motor on the burner unit resulted in more economical use of electric current. It is also apparent that the electrical input to the burner motor, which included the energy required for the spark ignition process, was sufficiently large to be considered as an important item in determining the overall cost of operation of the heating plant.

With the method of thermostatic control used in these tests, the times at which the fan operated practically coincided with the times at which the burner operated, or the frequency of operation was nearly the same for both fan and burner. The total time of fan operation was in most cases slightly greater than the total time of burner operation, but the differences were so small that for all practical purposes the data presented in the second set of curves in Figs. 11a and 11b apply equally well to the fan and the burner. The electrical input to the fan motor as shown by the bottom set of curves in Figs. 11a and 11b was smaller for the higher rates of oil input.

A comparison of total operating costs for an *average* heating day are of interest. For Urbana, Illinois, the average outdoor temperature during the heating season is 38 F. Hence for an indoor temperature of 71 F, the value of the average temperature difference between indoors and outdoors is 33 F. The comparisons based on this temperature difference may be regarded as indicative of the results to be secured from the entire heating season. For a day in which the indoor-outdoor temperature difference was 33 F the combined electrical energy inputs to the fan motor and burner motor were 4.4 kwhr and 3.6 kwhr for oil input rates of 9.5-lb and 13.0-lb per hour respectively. That is, the increase in fuel oil consumption, amounting to 6 lb of oil per 24 hours which accompanied the use of the higher rate of oil input to the furnace, was partly offset by a decrease in consumption of electrical energy amounting to 0.8 kwhr. Based on unit costs of 7 cents per gallon for fuel oil and of 3.1 cents per kwhr for electrical energy, the increase in fuel cost amounting to 5.7 cents per day and the decrease in electrical cost amounting to 2.5 cents per day resulted in a net increase in cost of 3.2 cents per day. For conditions under which the unit electrical cost is much higher than 3.1 cents per kwhr, the net increase in cost might become negligible.

It is apparent from this study that seasonal operating costs of an oil-fired, forced-air heating system based only on the cost of fuel may be misleading, and that the total cost of operation which includes electrical costs should be considered. In most installations the minimum capacity limitations of the burner and the desirability of maintaining ample reserve capacity for sudden load demands make it necessary to provide an oil input rate that is somewhat in excess of the maximum heating demands. However, it may be concluded from these tests that from the standpoint of total operating cost the most economical operation would be obtained with the use of the minimum rate of oil input that is feasible for the installation.

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Comparison of Conversion and Oil-Burning Units

The tests which were made to determine the comparative performance characteristics of the conversion unit and the oil-burning furnace under actual service conditions were conducted under identical conditions of operation. The results obtained for an oil input rate of 13.0-lb per hour with the conversion unit were presented in Figs. 10b and 11b and have been transferred to Figs. 12 and 13 for purposes of comparison with the results obtained with the

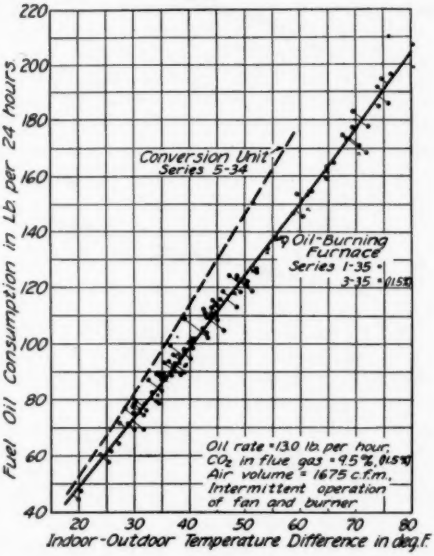

FIG. 12. FUEL CONSUMPTION CURVES FOR TWO OIL FURNACES

oil-burning furnace. The curves representing the results for the latter furnace are shown as full lines.

It may be noted from the fuel consumption curves shown in Fig. 12 that the fuel requirements, for a day in which the indoor-outdoor temperature difference was 33 F, were 94-lb for the conversion unit and 82.5-lb for the oil-burning furnace. Thus the fuel requirements for an average day were approximately 14 per cent greater for the conversion unit than they were for the oil-burning furnace. For an indoor-outdoor temperature difference of 55 F the difference was greater, and amounted to approximately 19 per cent.

Slight differences in operating characteristics were also obtained, as are shown by the curves in Fig. 13, for burner cycles, burner operation, burner motor input, and fan motor input. Very little difference may be observed in the number of cycles of burner operation per day for the two installations. However, both the number of hours of burner operation and the electrical energy input to the burner motor were greater for the conversion unit than for the

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oil-burning furnace. This could be accounted for by the fact that although the rate of oil input was the same in the two installations, the total fuel consumption was greater for the conversion unit than for the oil-burning furnace and hence the hours of burner operation were also greater.

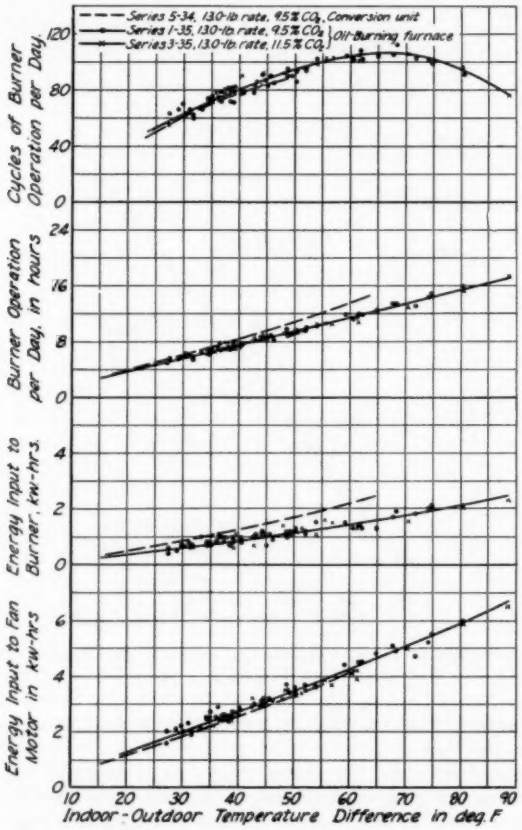
Series 5-34, 130-lb rate, 9.5% CO2, Conversion unit
Series 1-35, 130-lb rate, 9.5% CO2, Oil-Burning furnace
Series 3-35, 130-lb rate, 11.5% CO2
Cycles of Burner Operation per Day.
Burner Operation per Day, in hours
Energy Input to Burner, kw-hrs.
Energy Input to Fan Motor in kw-hrs
Indoor - Outdoor Temperature Difference in deg. F

FIG. 13. PERFORMANCE DATA FOR BURNER AND FAN OPERATION IN RESEARCH RESIDENCE, OIL FURNACE, FORCED-AIR HEATING SYSTEM. SEASON 1935-36, INST. 16

It may be observed that although the total time of fan operation per day was approximately 12 per cent less for the oil-burning furnace than for the conversion unit, the total electrical input to the fan was approximately 8 per cent greater. This difference can be accounted for by the fact that, due to the greater resistance to air flow in the case of the oil-burning furnace and also due to slight differences in motor characteristics, the rate of electrical input

to the fan motor for the oil-burning furnace was approximately 25 per cent greater than that to the fan motor for the conversion unit.

For a day in which the temperature difference between indoors and outdoors was 33 F, the combined electrical energy inputs to the fan and burner motors were 3.1 kwhr for the conversion unit and 2.9 kwhr for the oil-burning furnace, or a net difference of 0.2 kwhr per day. The difference was greater for days having greater heat demand. The operation of the oil-burning furnace resulted, therefore, in a net reduction in fuel oil of 11.5 lb and a net reduction in electrical energy of 0.2 kwhr per day. Based on unit costs of 7 cents per gallon for fuel oil and 3.1 cents per kwhr for electrical energy, these reductions were equivalent to 10.9 cents per day for fuel and 0.6 cents per day for electrical energy, or a total of 11.5 cents per day. For an average heating season consisting of 210 days this is equivalent to a difference in net operating costs of approximately $24. It may be concluded from these tests that under identical conditions of operation, the best economy was secured with the furnace which was equipped with ample heating surface and which was designed specifically for oil burning.

Variations in CO_2 in Flue Gas

In order to determine the operating characteristics of the oil-burning furnace when the fuel was burned under the best attainable combustion conditions, the air inlet to the burner was adjusted until a value of 11.5 per cent CO_2 in the flue gas was obtained. It was not possible to increase the CO_2 content much beyond this value without obtaining some indication of unburned combustibles. Otherwise, the operating conditions were maintained the same as in the previous tests; namely with 1675 cfm air circulated, 13.0-lb oil rate, and 0.02 in. draft in the combustion chamber. The results obtained from this test are shown in Figs. 12 and 13 in which the points are designated as belonging to Series 3-35. In general, no appreciable difference, either in fuel consumption or plant operation, was obtained when the CO_2 content in the flue gas was increased from 9.5 per cent to 11.5 per cent. However, when the heat demands of the house were large, there was a slight tendency towards an improvement in conditions with the higher CO_2 content. These results are in substantial agreement with those obtained from the tests made under conditions of continuous operation, as shown in Fig. 8. For example, for an oil input rate of 12.84 lb per hour, the increase in bonnet efficiency was approximately 1.5 per cent when the CO_2 content was increased from 9.5 per cent to 11.5 per cent. For all practical purposes the results obtained with the two conditions of combustion may be regarded as identical.

These results serve to emphasize the statement previously made that an oil-burning furnace which is provided with sufficient effective heating surface should perform satisfactorily notwithstanding fairly wide deviations in the CO_2 content of the flue gas.

Overall House Efficiency

In order to compare the fuel consumptions with oil and anthracite, the fuel quantities shown in Fig. 12 were reduced to terms of heat input to the furnace, in millions of Btu per 24 hours, and were plotted as shown in Fig. 14. The

results for the conversion unit (Curve No. 1) and for the oil-burning furnace (Curve No. 2) were obtained with an oil input rate of 13.0-lb per hour, a combustion condition of 9.5 per cent CO_2 and an air circulation of 1675 cfm. It should be noted that these tests were based on intermittent operation of the fan and oil burner, whereas the tests on the hand-fired coal furnace (Curve No. 3) were based on the maintenance of continuous combustion under which conditions the automatic controls varied with the combustion rate so that it approximately corresponded with the heating demands of the house. In all three cases the operation of the heating plant was controlled by means of sensitive thermostatic controls.

From analyses of the flue gases when anthracite was used as a fuel, the heat lost in the gases escaping from the chimney, which represented heat furnished by the fuel that was not ultimately available for heating the house, was determined as approximately 10 per cent of the total heat input to the furnace. The actual heat loss from the house (Curve No. 4) was then derived from the experimental results for heat input obtained with anthracite as a fuel (Curve No. 3). By using these derived values of the actual heat loss from the house in connection with the fuel consumption curves for oil, the overall house efficiency could be calculated for the cases in which oil was used as a fuel in the conversion unit and in the oil-burning furnace. These curves representing overall house efficiency are shown in the lower part of Fig. 14. It may be noted that the values of overall house efficiency determined indirectly in this manner for the oil-burning furnace agreed reasonably well with the calculated values of overall house efficiency determined directly from an analysis of the flue gas losses during the periods of burner operation.

For the same weather conditions, the values of the heat input were less and the values of the overall house efficiency were greater for anthracite than for oil. The most obvious explanation for this difference is that the combustion efficiency for oil combustion was considerably less than that for anthracite. This most obvious explanation, however, was not satisfactory since there was no reason to believe that a material difference existed either in the combustion efficiency, or that the flue gas losses were greater in the one case than in the other. In order to make an independent analysis of the factors affecting the utilization of heat by the furnace, the weight of the products of combustion and the flue gas losses were calculated from the flue gas analyses and the temperatures of the flue gas, observations of which were made under conditions of continuous operation of the burner and fan. These results are shown in Fig. 15a plotted against the heat input to the furnace.

The flue loss for any fuel may be considered as a composite value which is dependent on both the temperature and the weight of the flue gas. A comparison of the curves representing flue gas temperature in Fig. 15a shows that the flue gas temperatures were lower for oil, when it was burned in the oil-burning furnace, than for anthracite. This indicates that the flue gas temperature alone is not a reliable index of combustion efficiency when comparing different fuels. The products of combustion may be considered as being composed of dry gas and water vapor. The curves in Fig. 15a indicate that the weight of the dry gas was approximately the same for oil and for anthracite, but that the weight of water vapor was greater for oil than for anthracite, principally on account of the greater percentage of hydrogen in the former fuel.

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The combined effect of the flue gas temperature and the weight of flue gas determined the flue gas loss for various rates of heat input to the furnace. In this connection it may be observed, that it would be possible to still further reduce the flue gas temperatures by the addition of effective heating surface

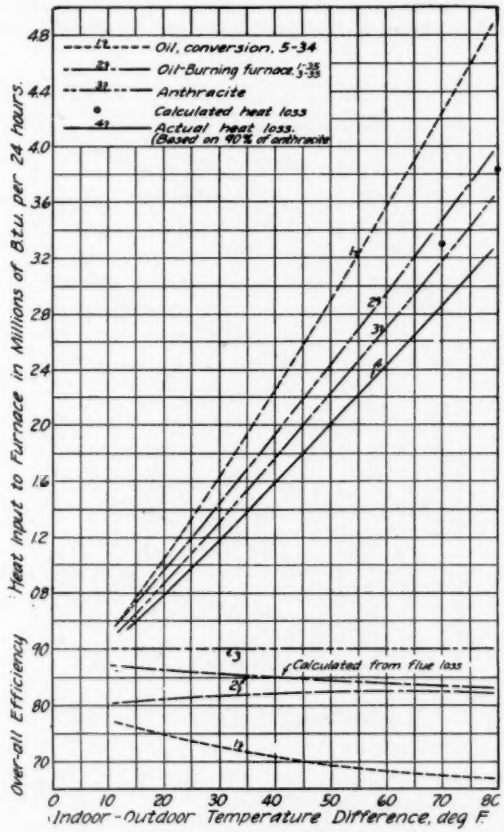

FIG. 14. HEAT INPUTS TO RESEARCH RESIDENCE

to the furnace. This would result in a smaller flue gas loss for all combustion rates, and hence in an increased efficiency. As indicated by the top set of curves in Fig. 15a the flue gas loss for a given value of heat input was approximately the same for oil (oil-burning furnace) and for anthracite. In other words, the combustion efficiencies in the two cases were approximately the same. Hence, some explanation involving factors in addition to combus-

tion efficiency must be sought to account for the differences obtained in the heat input for a given difference in temperature between indoors and outdoors.

In the case of anthracite, the automatic control varied the rate of combustion so that it approximately corresponded with the heating demands of the house; namely with the difference in temperature between indoors and outdoors. Hence the combustion efficiency, as determined by the rate of combustion and therefore the heat input to the furnace, would be directly reflected in the

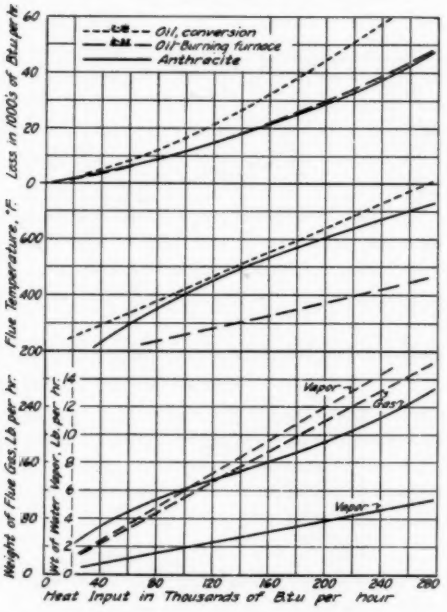

FIG. 15a. ANALYSIS OF FLUE GAS LOSSES

curves of heat input to the house, Fig. 14, which are based on the temperature difference between indoors and outdoors.

However, in the case of oil, the rate of oil input, or the hourly heat release, was constant for all weather conditions, and control was obtained by adjustments in the length of time that the burner was allowed to operate. Therefore, the combustion efficiency would be dependent on this rate of heat release and not on the temperature difference between indoors and outdoors. Furthermore, the net flue gas loss for a 24-hour period, during which the burner was operated intermittently, would be determined not only by the combustion efficiency but also by the total length of time during which the burner was actually in operation. The oil burner was so adjusted that the hourly heat release was somewhat greater than the hourly heat loss from the house on the maximum

day. The hourly heat release in the case of anthracite, however, was just sufficient to offset the hourly heat loss from the house. Hence even on the maximum day the hourly heat release in the case of the oil-burning furnace was greater than that for anthracite, and the curves in Fig. 15a show that, although the combustion efficiencies for the oil-burning furnace and the anthracite were the same at the same heat release, during the actual periods of operation the flue gas loss was greater and therefore the combustion efficiency was

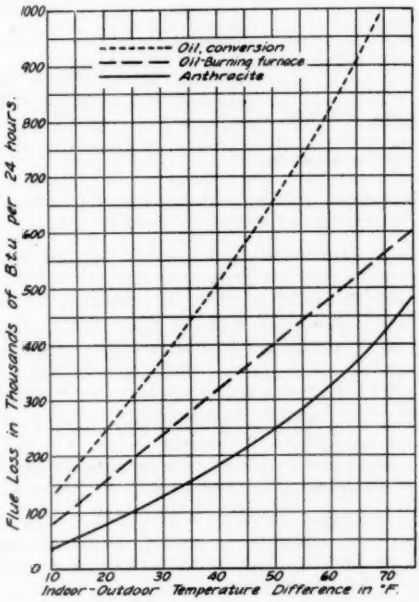

FIG. 15b. CALCULATED TOTAL FLUE LOSS

less for the oil-burning furnace than it was for the anthracite. Since the rate of oil input remained the same for all weather conditions it is therefore evident that the oil-burning furnace always operated at a less combustion efficiency than the anthracite during the periods of actual operation. The net flue gas loss during a 24-hour period would be determined by the product of the hourly flue gas loss during the actual operating period and the total hours of burner operation during the day. The calculated values of net flue gas loss shown in Fig. 15b were derived from the data presented in Fig. 15a and are consistent with the experimental data for the heat inputs shown in Fig. 14 in that the curves representing higher flue gas losses in Fig. 15 correspond in position with those representing the higher heat inputs in Fig. 14. It is the total amount of the net flue gas loss over a given period of time

TABLE 3—RELATIVE FUEL COSTS FOR AN AVERAGE HEATING DAY
(*See Fig. 14*)
(*Based on prices in Urbana, Illinois*)

	ANTHRACITE	OIL-BURNING FURNACE	OIL CONVERSION UNIT
Unit cost of fuel	$16.50 per ton	7 cents per gallon	
Calorific value	13,200 Btu per lb	144,200 Btu per gallon (19,500 Btu per lb)	
Cost per therm[a] in cents	6.25	4.85	
Heat input for 33 F temp. diff. per 24 hours	1,470,000 Btu	1,600,000 Btu	1,830,000 Btu
	14.7 therms	16.0 therms	18.3 therms
Fuel cost for average day, in dollars	0.92	0.776	0.888

[a] One therm is equivalent to 100,000 Btu.

rather than the combustion efficiency alone, that determines the relative position of the curves of heat input shown in Fig. 14.

This analysis of the factors affecting the net flue gas loss indicates that in order to secure minimum flue gas losses from an oil burner, the flue gas temperatures and the weight of the products of combustion should be maintained at a minimum value. This condition can be most nearly approached by adjusting the rate of oil input so that the heat release just offsets the heat loss from the house in the most severe weather. This analysis also suggests that if the rate of oil combustion could be varied to conform at all times with the heating demands of the house, an overall house efficiency equal to that obtained with anthracite should be possible of attainment. Since the combustion efficiency is affected by the amount and location of the heating surfaces, the necessity of providing adequate and effective heating surface is apparent.

A comparison of fuel costs, based on prices paid in Urbana, Illinois, for an average heating day is of interest. From Fig. 14 for an indoor-outdoor temperature difference of 33 F the values of heat input per day were 1,830,000 Btu, 1,600,000 Btu and 1,470,000 Btu for the conversion unit, the oil-burning furnace, and anthracite furnace, respectively. Based on unit fuel costs of 7.0 cents per gallon for oil having a calorific value of 144,200 Btu per gallon, the cost per therm is 4.85 cents. For anthracite costing $16.50 per ton, having a calorific value of 13,200 Btu per pound the cost per therm is 6.25 cents. The daily fuel costs based on these unit costs are given in Table 3.

It may be noted that the fuel costs for an average heating day were 77.6 cents, 88.8 cents, and 92.0 cents for the oil-burning furnace, the oil conversion unit, and anthracite, respectively. These cost comparisons, of course, are valid only for the unit cost of fuels listed in the table and apply to prices paid in Urbana, Illinois.

CONCLUSIONS

The following conclusions may be drawn as applying to the Research Residence and the conditions under which the tests were conducted.

(1) The capacity and efficiency of oil-burning, warm-air furnaces are dependent upon seven factors:

1. Type of burner
2. Rate of oil input
3. Percentage of CO_2 in flue gas
4. Condition of draft in combustion chamber
5. Amount and location of heating surfaces
6. Size and shape of combustion space
7. Quantity of circulating air.

(2) With a constant percentage of CO_2 in the flue gas, and a constant rate of oil input, the efficiency and capacity of the furnaces using oil as a fuel were increased when the quantity of air circulated was increased.

(3) With a constant quantity of air circulated and a fixed rate of oil input to the furnaces using oil, the capacity and efficiency of the unit were not materially increased when the CO_2 content in the flue gas was increased from 9.5 per cent to 11.5 per cent.

If furnaces using oil as a fuel are provided with sufficient effective heating surface for a given rate of oil input, so that the temperature of the leaving flue gases will be low, then the unit may be expected to perform satisfactorily under diverse conditions of installation in the field, even with fairly wide deviations in the adjustment of the fuel-air ratio for the oil burner.

(4) Under normal service conditions, the operation of the burner for relatively short periods, during which a high rate of combustion was maintained, was not as good from the standpoint of fuel economy as the operation of the burner for longer periods, during which a lower rate of combustion was maintained.

(5) Seasonal operating costs of an oil-fired, forced-air heating system based only on the cost of fuel may be misleading, since the total cost of operation including the electrical costs should be considered.

(6) For an average heating day, or a day on which the indoor-outdoor temperature difference was 33 F, the requirements of the oil-burning furnace as compared with the requirements of the conversion unit showed a net reduction in fuel oil of 11.5 lb per day and a net reduction in electrical energy of 0.2 kwhr per day.

(7) For the same weather conditions, the heat inputs to the furnace were less and the overall efficiencies were greater for anthracite than for oil.

The temperature of the flue gases alone is not a reliable index of combustion efficiency when comparing different fuels. In fact, the total amount of the net flue gas loss during a 24-hour period, rather than the combustion efficiency alone, determines the magnitude of the heat input required to maintain house temperatures.

(8) Based on unit costs of 7 cents per gallon for fuel oil and $16.50 per ton for anthracite, the fuel costs for an average heating day, or a day in which the indoor-outdoor temperature difference was 33 F, were 77.6 cents, 88.8 cents, and 92.0 cents for the oil-burning furnace, the oil conversion unit, and the furnace burning anthracite, respectively.

ACKNOWLEDGMENTS

The results presented in this paper were obtained in connection with the investigation of warm-air furnaces and heating systems in the Research Residence,[6] Fig. 1, at the University of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, Dean of the College of Engineering, is the director, in the Department of Mechanical Engineering of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the head. This investigation is a cooperative project sponsored jointly by the *National Warm Air Heating and Air Conditioning Association* and the Engineering Experiment Station. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is also due to J. S. Cunningham, former Research Assistant, for services rendered in connection with the investigation, and to the various manufacturers who cooperated by furnishing instruments and equipment.

[6] The Research Residence in Urbana, Ill., was built, furnished, and completely equipped specifically for research work in warm-air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

DISCUSSION

L. C. PRICE:[7] I would like to refer to Fig. 9, showing the plot of total stack loss against CO_2. Professor Konzo pointed out that when a constant stack temperature is maintained and a change of percentage of CO_2 goes from 12 to 10, there is not much change in the total stack loss, but he also pointed out, or certainly Mr. Cross did, that when you change the CO_2 from 12 to 10, with a constant rate of fuel supply, your stack temperature no longer remains the same. Now, to show the combined effect of the change in stack temperature and the change in CO_2, both of which result simultaneously from a change in air supply, a plot of stack loss versus air fuel ratio would be very interesting. I would like to ask Professor Konzo if he made any of those calculations.

PROF. S. KONZO: We have not made any such calculations that correlate the stack loss against the air fuel ratio. We will take his suggestion under advisement, although I am not sure that we can supply him with the desired information.

E. B. LANGENBERG: I should like to comment on the two papers. Surely, right now, the air conditioning industry is facing a crisis. The public is commencing to accept the subject of air conditioning, and expecting a great deal more than is supplied them. There is one thing we have to watch out for and that is to see that we do not exaggerate or claim we can do more than we *can* do, and we should take the time to do some research work on our own jobs.

In the past 45 years, our company has put in about 100,000 plants in the City of St. Louis. We have accumulated some experience but I regret to say, the amount of real analysis we made of those plants was negligible. From now on, the more data of this character that we can bring here and discuss, the better off we will be.

There are many variables in these new systems. If one figures a combination heating and cooling system, he will run up to 101 very easily. Any change in these variables will affect the system. We must have facts to base our calculations on, so when we give the public an air conditioning system it will be one.

Filtration is another thing. Plants are set up with one or two filters and the complaints that are being made are that the degree of filtration is no good, and they are clogging up, creating additional resistance, and motors are burning up.

We cannot afford, in this industry, to jeopardize the future by trying to make it such a commercial proposition that we are out hunting for the dollar today, and forgetting the dollar tomorrow. I would suggest to every manufacturer in the room, or to you men who represent the manufacturers, that you go back to your plants and suggest to the boss that when you put in the next one hundred plants, you analyze along the lines of these two papers, and any other additional information you can get, find out what you can do in a house, and one of these days, some of us, or collectively, we are going to find the correct idea that will balance all of these things so when you put a plant in a man's house, it will stay put and will give him a hundred dollar value for his hundred dollars.

Keep those things in mind; our future and the future of the whole air conditioning industry depend on them. The research work at the University of Illinois, which the manufacturers are backing, is an asset that a great many are losing sight of. I have men come into the office and try to tell me certain things about air, and I say, "Have you gone through the bulletins of the University of Illinois?" If they haven't, I say, "I don't think you know anything about air if you haven't studied those particular bulletins." We have spent close to $100,000 for that research up to date and while we do not claim to be perfect, we are learning rapidly.

[7] Research Engr., University of Arkansas, Fayetteville, Ark.

No. 1058

CONDENSATE AND AIR REMOVAL RATES IN A VACUUM HEATING SYSTEM

By D. W. NELSON* (*MEMBER*), MADISON, WIS.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the University of Wisconsin.

THIS paper reports the results of a study of the air and condensate loads in the heating system returns of the Mechanical Engineering Building at the University of Wisconsin, an exterior view of which is shown in Fig. 1. A single story portion between the two wings houses the heating and ventilating laboratory which has been described in a previous paper.[1] The present study is on the returns from the vacuum system of the entire building, including this single story section used in the previously reported tests on the three heating systems. Observations were made during the heating seasons of 1934-35 and 1935-36.

DESCRIPTION OF HEATING SYSTEM

The main part of the building has a total of 26,244 sq ft of equivalent direct radiation, made up as follows: 5709 sq ft of cast-iron radiation in classrooms and offices; 1421 sq ft of wall radiation in stairways and corridors; 7566 sq ft of EDR in 28 unit heaters in various laboratories; 4128 sq ft of EDR in 8 unit ventilators in the auditorium, conference and wash rooms; and 7420 sq ft EDR in cast-iron blast coils for the classroom ventilating system.

The classroom and office radiation is on two temperature control which lowers the temperature setting to 50 F from 5 p.m. to 5 a.m. The radiation in stairways and halls is manually controlled, as is the starting and stopping of the central fan and unit heaters. The fan system, unit heaters and ventilators are all under single or continuous thermostatic control.

The heating system of the saw-tooth single-story section consisted originally of cast-iron radiation of the wall type placed along the walls under windows and below skylights. The system has been altered to the extent of adding seven suspended unit heaters, and two floor unit heaters with short branches

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[1] Tests of Three Heating Systems in an Industrial Type of Building, by G. L. Larson, D. W. Nelson and John James, A.S.H.V.E., TRANSACTIONS, Vol. 41, 1935, p. 185.

Presented at the 43rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, St. Louis, Mo., January, 1937.

from the original system. These are thermostatically controlled as three separate systems, any one of which may be operated. The cast-iron direct radiation is about 4430 sq ft, the suspended unit heaters 4318 sq ft EDR and the floor unit heaters 3790 sq ft EDR. This is a total of 12,538 sq ft connected to the system with about one-third in active use at any one time. The remainder is open to the returns at all times and acts as a reservoir for air and contributes to the air leakage into the system to a certain extent.

The total connected equivalent direct radiation based on a 240 Btu emission factor is 38,782 sq ft, with about 30,500 sq ft in active use at any one time. The internal volume of the supply piping is about 590 cu ft, and that of the returns 74 cu ft. Cast-iron direct radiation internal volume is 186 cu ft, and for all other units including the blast coils it is estimated to be 22 cu ft. This is a total internal volume of the system of 872 cu ft.

The supply of steam is through a tunnel from the central heating plant where it is generated at about 150 lb per sq in. gage and reduced to about 10 lb per sq

FIG. 1. PHOTOGRAPH OF MECHANICAL ENGINEERING BUILDING AT THE UNIVERSITY OF WISCONSIN

in. gage before entering the tunnel mains. A further reduction in pressure to about 3 lb per sq in. gage takes place at the building entrance by means of a reducing valve. The steam laboratory located in the east wing is supplied with steam at boiler pressure for engine tests, and the condensation loss in this pipe line enters the return line of the heating system. The heat loss from this main, of course, is useful heat during the heating season.

The returns terminate in a 4-in. pipe at the vacuum pump located in the basement of the front section of the building. The condensate is pumped from here through 300 feet of 2 in. pipe to the tunnel entrance, where it passes through two condensation meters in parallel, each having a capacity of 12,000 lb per hour. It then is pumped back to the central heating plant by a direct acting steam pump. The vacuum pump was rated by the manufacturer for a system of 65,000 sq ft and a capacity of 90 gpm and 34 cfm at a 10 in. vacuum.

TEST ARRANGEMENT

An arrangement used in observing the air and condensate loads is shown in Fig. 2. The air discharge was piped into a pressure chamber made of a 30 in. length of 6 in. pipe from which it reached the atmosphere through a 1 in.

thin-plate sharp-edge orifice. The pressure on the orifice lifted a float attached to a rheostat. This varied the voltage to a recording volt meter which was supplied with a constant source of current through two six-volt storage batteries in series which were continuously charged by a trickle charger. The recording chart was moved at the rate of 3 ft per day and gave a record of pressures across the orifice and running periods for the vacuum pump. The system also was equipped with a recording pressure gage on the end of the steam main and a recording vacuum gage on the end of the return main, both of these being near the vacuum pump. In addition, manometers were installed to check these pressures and the pressure on the orifice.

For continuous tests of steady operation over a period of days, the condensation meters were read at frequent intervals and a calibration factor secured between running times of the pump as shown on the voltage recorder

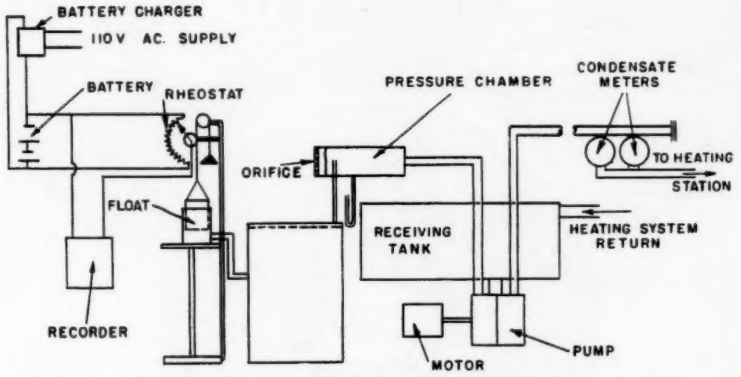

FIG. 2. ARRANGEMENT OF TEST EQUIPMENT FOR CONDENSATE AND AIR REMOVAL TESTS

and condensation as shown on the condensation meter dials to apply at times inconvenient for observations, such as at night. This was found to give accurate checks so that a special recording device was not devised for the condensation meters.

During observations of peak conditions occurring during the first hours when steam was turned into the cold system, readings of all values were taken directly at 1 min intervals by five observers; three located in the pump room and two in the tunnel entrance.

OPERATION OF VACUUM PUMP

The three vacuum charts in Fig. 3 give a general picture of the operation of the pump. The chart at the upper right, starting at 5 p.m., April 5, shows the behavior in normal operation. This chart shows that during the night, operation on vacuum trip occurred every 1¼ to 1¾ hours with rather frequent

short periods of operation to remove condensate on float control in between these longer runs to remove air. The vacuum trip cut in at about 3 in. Hg and out at about 8 in. Hg, during which time any condensate in the receiving tank was discharged. The running times on float control were found to average very closely to 20 sec in duration during the past season. During the previous

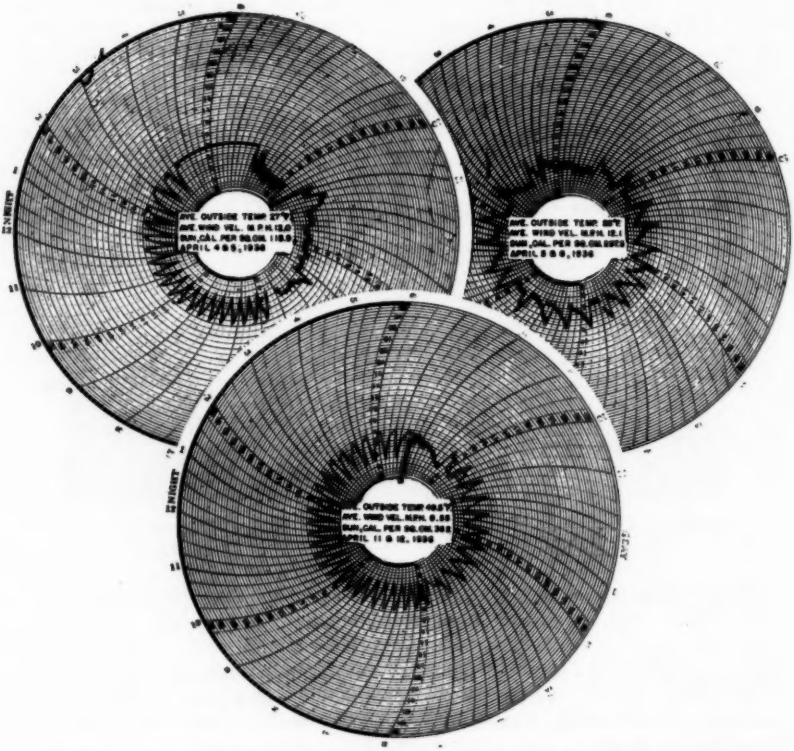

FIG. 3. VACUUM RECORDER CHARTS FOR VARIOUS OPERATING CONDITIONS OF THE SYSTEM

season the float was adjusted to a shorter travel, which resulted in operating periods of 12 sec. The amount of condensate removed during each operating period on float control would be the receiving tank capacity between the cut in and out levels plus any additional amount entering during this short period of operation.

The chart for April 5-6 in Fig. 3 shows a warming up period of almost 3 hours with many short float controlled runs following the change to day setting of the two temperature controlled radiation in the classrooms, offices and in the

heating and ventilating laboratory. The operation on vacuum trip occurred at intervals of about 1 hour during the day with four or more operations on float control in between for condensate removal. Of course, air was also removed during the float controlled periods, and if this type of operation were frequent enough to maintain the vacuum, no vacuum trips would occur. If the air removed was in excess of air in the system, the vacuum would rise above the normal range and the vacuum breaker would introduce air from the atmosphere. This condition was not observed at any time, although an erratic condition was found on April 5 as shown on the upper left chart of Fig. 3. The main valve on the low pressure supply main into the building

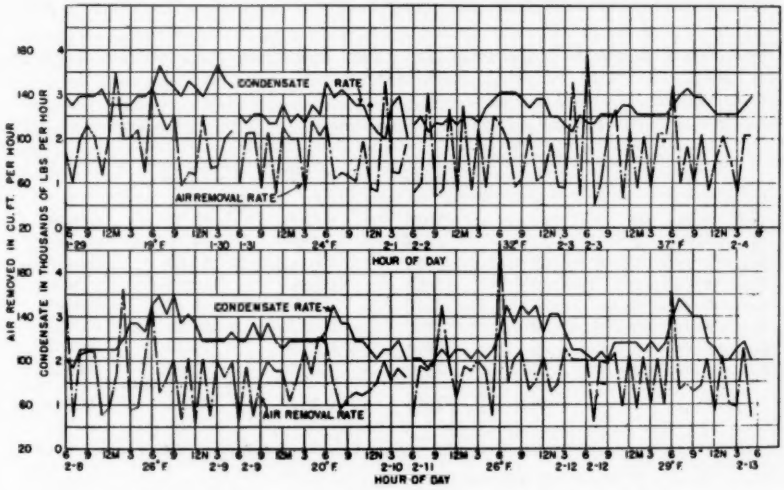

FIG. 4. CONDENSATE AND AIR REMOVAL RATES FOR EIGHT TYPICAL DAYS

was closed the night before at 5 p.m. and at 8 a.m. the valve was opened. After a short period of operation, the pump operated on cycles of only a few seconds' duration and yet the vacuum in the receiver tank increased to 14 in. Hg, which caused the vacuum breaker to introduce room air. This difficulty was found to be due to a clogged strainer, and when it was cleaned shortly after 10 a.m. the pump resumed normal operation. The two upper charts are for consecutive days with similar weather conditions and indicate a difference in air leakage, depending upon whether the steam was on or off. On the night of April 5 the steam was on the mains and in the various heating units except as prevented by the lower temperature settings for the two temperature controlled units. The pump operation was similar to a typical day operation with short float controlled runs in between vacuum controlled runs. On the night of April 4 the steam was shut off from the entire heating system. As the system and building cooled down, the pump operated at increasingly longer intervals in order to remove the leakage air to maintain the vacuum between

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3 and 8 in. Hg. From shortly after 4 a.m. to 8 a.m. the pump ran continuously without appproaching the cut-off vacuum. This shows a decided increase in air leakage as the system cooled down, although a survey of the building did not disclose any noticeable air leaks. The single story section was built 15 years ago, whereas the main building system was installed in 1930. It is known that the wall radiators in the older section, which have as many as 28 sections, are not particularly tight. However, unit heaters were being used for heating during this period; consequently, these radiators were without steam during both of these nights. The air leakage varied during these nights when the steam was off for some unknown reason. The lower chart of Fig. 3 indicates less air leakage than in the upper left chart as shown by the fact that it was not necessary for the pump to run continuously. The steam was

TABLE 1—CONDENSATE, AIR REMOVAL RATES AND LOAD FACTORS FOR NORMAL OPERATION

DATE	AVERAGE OUTSIDE TEMPERATURE DEG F	CONDENSATE		AIR REMOVAL	
		Rate in Lb per Hour	Load Factor[a]	Rate in Cu Ft per Hour	Load Factor[a]
Jan. 29–Jan. 30...	19	3055	0.0673	97.20	0.0481
Jan. 31–Feb. 1...	24	2628	0.0581	83.75	0.0415
Feb. 2–Feb. 3...	32	2552	0.0567	82.80	0.0410
Feb. 3–Feb. 4...	37	2669	0.0593	88.80	0.0440
Feb. 8–Feb. 9...	26	2610	0.0580	88.40	0.0438
Feb. 9–Feb. 10...	20	2490	0.0554	83.20	0.0412
Feb. 11–Feb. 12...	26	2475	0.0550	94.20	0.0461
Feb. 12–Feb. 13...	29	2410	0.0536	82.60	0.0409
Average.....	26.6	2614	0.0579	87.62	0.0433

[a] Based on a condensate capacity of 90 gpm and an air capacity of 34 cfm. Values not corrected for vacuum variations.

quickly turned on shortly before 7:30 a.m. with a resulting fall in vacuum to a positive pressure in the returns followed by frequent operation on float control until 9:30 a.m. This condition will be discussed in connection with Fig. 8. Operation during the day was at ¾ hour intervals on vacuum trips with two float controlled runs in between these longer runs. The weather on this day was considerably milder than on the other two days shown in Fig. 3.

OBSERVATIONS UNDER NORMAL OPERATION

The hourly condensate and air removal rates for the system on 8 typical days are plotted in Fig. 4. In general, a peak of air removal occurs after 5 a.m. each day when the radiation is shifted from night to day thermostatic setting. The highest of these peaks occurred on February 12 and amounted to 210 cu ft per hour. In each case the air removal peak leads the condensate removal peak so that the two do not coincide. The highest condensate peak occurred on January 30 and amounted to 3600 lb per hour. The average condensate and air rates for each of these days are tabulated in Table 1, together with

the ratio of these rates to the rated capacities of the pump. For the 8 days the average air removal rate was 4.3 per cent of the rated capacity and the similar comparison on condensate was 5.8 per cent.

The daily condensate and air removal rates for 12 typical days are plotted in Fig. 5. This chart indicates that the air removal rate was not influenced by the outside temperature but remained substantially constant over a range from 15 to 37 F. As discussed in Fig. 3, turning the steam off from the system materially increased the leakage. Since the radiators remain cold for increasingly longer periods under thermostatic control as the outside temperature becomes higher, the air removed might be expected to increase, giving an upward turn to the curve if extended to the right.

The system was designed for an outside temperature of −15 F. The condensate curve when extended indicates a steam consumption of 120,000 lb per day under this design condition. This is equivalent to 20,000 sq ft of EDR as compared to a normal capacity of the system of 30,500 sq ft. Since the fan system is used only during the day time, this condition and the lowered thermostatic setting on part of the system at night time would account for this difference.

PEAK LOAD OBSERVATIONS

The maximum condensate and air removal rates naturally occur when steam is turned into a cold system. In order to study these peak load conditions, the steam was turned off at 5 p.m. and on again early the next morning. Typical results are shown in Figs. 6, 7 and 8, which give the condensate and air removal rates, the vacuum in the return line, the differential pressure from the steam line to the return line and the periods of pump operation. All readings were taken at 1 min intervals, but for satisfactory plotting 3 min periods were used.

The gate valve in the 8 in. line at the tunnel entrance was opened over a period of 30 min for the test represented in Fig. 6. No attempt was made to calibrate the effective opening of the valve against turns of the hand wheel. Also the reducing valve, which was set to maintain 3 lb per sq in. on the building system, may have influenced the effective opening. The peak for air removal, which was 1825 cu ft per hour, was reached in 12 to 15 min. The pump ran continuously on vacuum control for 27 min. Considering the initial clearing of the air in the system extending until the pump stopped at 57 min, approximately 900 cu ft were removed, which is about equal to the internal volume of the system. Leakage under vacuum during this peak period would tend to raise the air removed above the internal volume of this system, but this would be counteracted by the air in the system being under less pressure therefore shrinking in volume when measured at atmospheric pressure.

The condensate peak of 9300 lb per hour came at 33 min, which is 18 min behind the air removal peak, and lasted for 33 min before fairly normal conditions existed. All of the pump operations except the first two were caused by float control, but the simultaneous removal of air was at such a rate as to maintain the vacuum quite constant around 7 in. Hg. The pump operated 30 per cent of the time in the 2 hours following the turning on of the steam. Weather conditions were uniform for the day preceding and the day of test, March 21

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and March 22, and averaged 42.5 F, with a sun intensity of 285 cal per square centimeter and a wind velocity of 9.5 mph.

The valve was presumably opened at the same rate of 30 min for full opening on March 8 for the test shown in Fig. 7. The outside temperature was 32 F, the sun effect 292 cal per square centimeter and wind velocity 7.5 mph for this day and the preceding day. The vacuum dropped to about 2 in. Hg at the air peak period which was 12 to 15 min after the time of starting to open the valve. At this time the differential pressure between the end of the supply

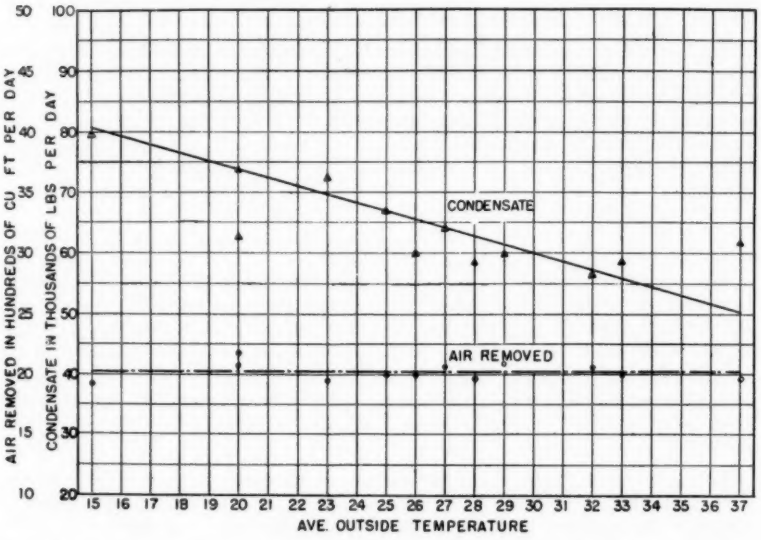

FIG. 5. RELATION BETWEEN CONDENSATE AND AIR REMOVAL RATES AND OUTSIDE TEMPERATURES

main and the pump end of the return dropped to zero, which meant that all traps were probably still open because steam had not reached them, and steam was not being supplied at a sufficient rate to build up a positive pressure in the steam mains. The air removal peak of 2175 cu ft per hour was reached in 15 min as was that shown in Fig. 6. Considering the peak lasting until 60 min, 1000 cu ft of air were removed. The condensate peak was high at 27 min after the starting of the opening of the steam valve, but the real 3 min peak of 12,000 lb per hour occurred at 36 min, which checks closely with the 33 min found in Fig. 6. The 1 min peak was 13,800 lb per hour. The ending of the peak period is rather indefinite, but probably was at about 60 to 80 min. All of the 25 pump operations in the first 2 hours, except the initial one of 27 min and the next two shorter ones, were under float control. The vacuum was maintained at a higher level than in the test shown in Fig. 6 and increased

somewhat towards the end of the 2-hour period due to the more frequent running of the pump. The condensate rate for steady operation shows about 4600 lb per hour in Fig. 7 and 3600 lb per hour in Fig. 6.

On several days the steam valve was opened quickly instead of over a period of 30 min as was done for the tests in Figs. 6 and 7. This resulted in a much higher air peak. The results of one of these tests taken on April 12 are shown in Fig. 8. The vacuum in the return dropped from above 8 in. Hg to ¾ in.

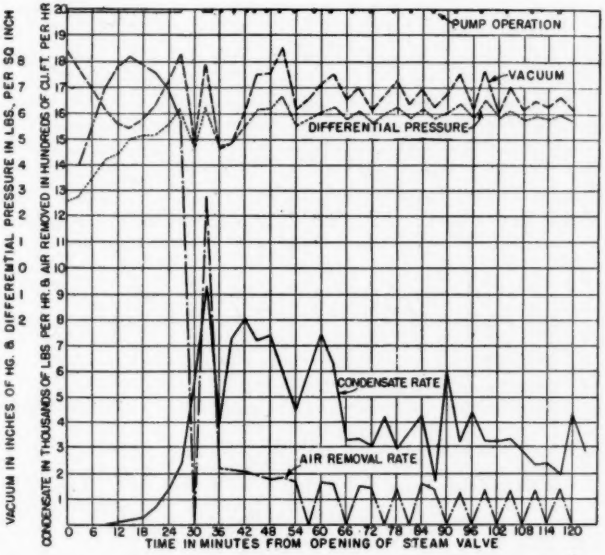
PUMP OPERATION
VACUUM
DIFFERENTIAL PRESSURE
CONDENSATE RATE
AIR REMOVAL RATE
VACUUM IN INCHES OF HG. & DIFFERENTIAL PRESSURE IN LBS. PER SQ INCH
CONDENSATE IN THOUSANDS OF LBS PER HR. & AIR REMOVED IN HUNDREDS OF CU.FT. PER HR
TIME IN MINUTES FROM OPENING OF STEAM VALVE

FIG. 6. PEAK LOAD TEST FOR MARCH 22, 1936, WITH SLOW OPENING OF STEAM VALVE

Hg in less than 1 min from the time of starting to open the valve. (The values of vacuums and differential pressures are instantaneous values at 3-min intervals and not averages for the 3-min periods, so Fig. 8 indicates the drop in vacuum as occurring in 3 min.) The pressure in the return at 6 min was 1½ lb per sq in. above atmospheric, indicating that the steam was materially assisting the pump in the removal of air. The 3 min peak of 2630 cu ft per hour was reached in 6 min and was largely over in about 15 min. The air removed in the peak was about 510 cu ft, which is about one-half of that removed in the 30 min opening tests. The fact that the pressure in the returns (as well as in heating units) was above atmospheric pressure for about one-half of this peak period could hardly account for the escape of air through leaks to the extent of one-half the internal volume of the system. On another test under similar conditions, this air peak was 680 cu ft total. On a somewhat

similar test at 11 a.m. on April 5 (see Fig. 3 upper left chart) the steam was turned into a warm system filled with air at atmospheric pressure with a resulting air removal peak of 400 cu ft. This test was made after the steam had been shut off for ½ hour and the return opened up to clean the pump strainer.

The condensate peak of 6800 lb per hour in Fig. 8 was arrived at 15 to 21 min after cracking the valve, and was substantially over at 39 min. This peak was low because of the mild weather as shown by averages of an outside tem-

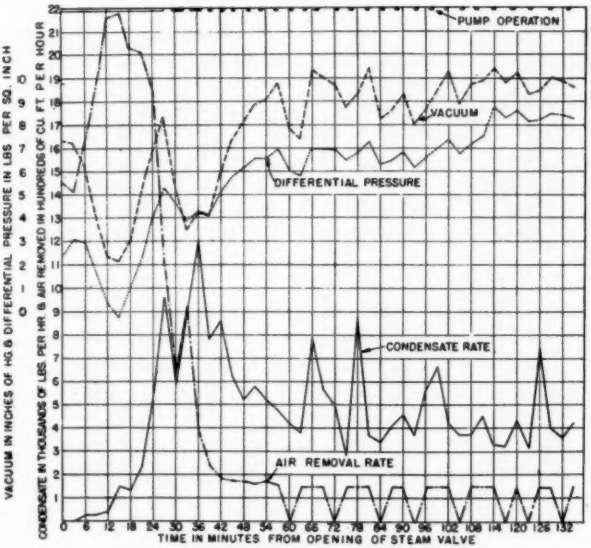
PUMP OPERATION
VACUUM
DIFFERENTIAL PRESSURE
CONDENSATE RATE
AIR REMOVAL RATE
VACUUM IN INCHES OF HG & DIFFERENTIAL PRESSURE IN LBS PER SQ. INCH
CONDENSATE IN THOUSANDS OF LBS PER HR. & AIR REMOVED IN HUNDREDS OF CU. FT. PER HOUR
TIME IN MINUTES FROM OPENING OF STEAM VALVE

FIG. 7. PEAK LOAD TEST FOR MARCH 8, 1936, WITH SLOW OPENING OF STEAM VALVE

perature of 49.5 F, sun effect of 382 cal per square centimeter and a wind velocity of 8.6 mph for this day and the day preceding. The lower chart in Fig. 3, which is for the period of this test, indicates less leakage into the system than at some other times as shown by the pump not running continuously during the time steam was shut off, as it did on several other similar occasions. After the air removal peak was reached, which was largely over in 12 min of operation, the pump operated entirely on float control until shortly after 2 hours of time had elapsed. During this period the vacuum steadily fell in spite of apparently less air leakage than on the other tests shown, due to the infrequent running to remove condensate and incidentally air. There were 17 starts of the pump in the 2-hour period. At 122 min of elapsed time the vacuum trip started the pump and an air removal peak with a rate of 1300 cu ft per hour for a 3-min period occurred.

A summary of the conditions and results of the peak load tests shown in

Figs. 6, 7 and 8 is given in Table 2. The ratio of the maximum condensate rate on the peak to the average rate after the peak was over and fairly normal conditions obtained was 2.56, 2.59 and 3.07. It would be expected that this ratio would be highest for the test with the most rapid opening of the valve, whereas it was not actually much higher. The maximum condensate rate depends not only on how rapidly steam is brought into the system but also on the temperature of the piping and heating units and their surroundings at that

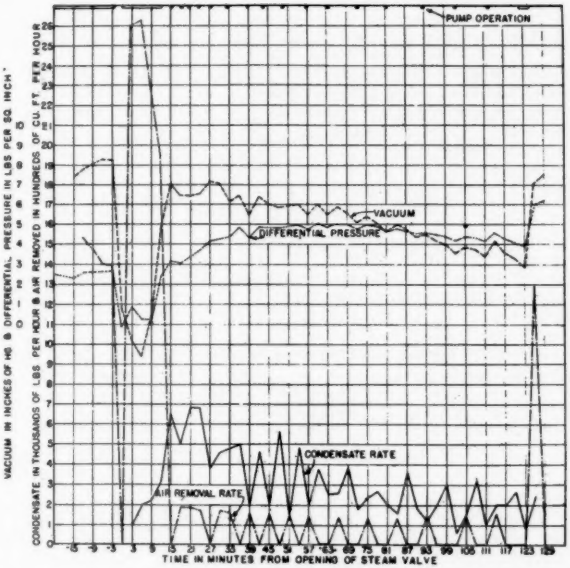
PUMP OPERATION
VACUUM
DIFFERENTIAL PRESSURE
CONDENSATE RATE
AIR REMOVAL RATE
VACUUM IN INCHES OF HG & DIFFERENTIAL PRESSURE IN LBS PER SQ INCH
CONDENSATE IN THOUSANDS OF LBS PER HOUR & AIR REMOVED IN HUNDREDS OF CU. FT. PER HOUR
TIME IN MINUTES FROM OPENING OF STEAM VALVE

FIG. 8. PEAK LOAD TEST FOR APRIL 12, 1936, WITH RAPID OPENING OF STEAM VALVE

time. The weather was the warmest (49.5 F) on the test with the most rapid opening of the valve, which would make this peak ratio lower than expected.

The location of the condensation meters at the tunnel entrance, removed from the vacuum pump by 300 ft of 2 in. discharge pipe, introduces a lag in the condensation readings. This was found to be about 30 sec by measuring the interval from the starting of the pump to the arrival of water to operate the meters. This discharge pipe is carried on the ceiling of the heating and ventilating laboratory some 15 ft in elevation above the level of the meter and, since the end of the line at the meter is equipped with an atmospheric vent, this pipe cannot act as a storage capacity. However, it discharges into a surge tank of 5000 lb capacity located directly before the meters, from which tank the condensation enters them under low gravity head. Since the two meters in parallel have a combined capacity under these conditions of 24,000 lb per hour and the highest peak observed was 12,000 lb per hour for a 3-min period,

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TABLE 2—SUMMARY OF PEAK LOAD TEST RESULTS

DATE		MARCH 22	MARCH 8	APRIL 12
DATA SHOWN IN FIGS.		6	7	8
Weather Conditions	Average Outside Temperature	42.5 F	32 F	49.5 F
	Average Wind Velocity, mph	9.5	7.5	8.6
	Sun Effect, Cal per Sq Cm	285	292	382
Approximate Time to Fully Open Valve, Minutes		30	30	4
Average Removal Rate for Normal Operation	Condensate, Lb per Hour	3630	4620	2215
	Air, Cu Ft per Hour	83.5	106.25	93
Time of Peak after Opening Valve, Minutes	Condensate	33	36	21
	Air	15	15	6
Removal Rate in Max. 3-min Peak	Condensate, Lb per Hour	9300	12000	6800
	Air, Cu Ft per Hour	1825	2175	2630
Ratio—Peak to Average	Condensate	2.56	2.59	3.07
	Air	21.8	20.48	28.3
Total Amount Removed during Peak Period	Condensate, Lb	4275	4310	2790
	Air, Cu Ft	897	1007	510
Total Pump Running Time for 2-Hour Period After Opening Valve, Hours		0.596	0.647	0.283
Pump Operation in Per Cent of Time for 2-Hour Test		29.8	32.4	14.3

it is considered that this is the true 3 min peak and that the lag in measuring condensation is of small consequence in the time relations of the air and condensation peaks. This lag likely accounts for some of the out-of-phase relation of condensation and air removal rates during near normal operation shown in Figs. 6, 7 and 8. The pump usually ran about 20 sec in any 3-min period and, when this occurred at the end of the period, the air would appear in that period and the condensation in the following 3-min period. No attempt was made to correct these minor differences.

A previous paper [2] reported to the Society indicates that the maximum load is encountered in a system at the time of maximum rate of condensation. The ratio of air to condensate at that time is therefore of importance and in those tests did not exceed 0.05 cu ft of air per pound of condensate when the system was operated at or above atmospheric pressure. In three tests under reduced pressures throughout the system this averaged 0.14 cu ft per pound.

In the present tests as reported in Figs. 6, 7 and 8 the returns were under approximately 8 in. vacuum at the start of the tests and a vacuum was maintained within the capacity of the pump under the peak load conditions. The vacuum extended into the supply mains up to the time of opening of the steam

[2] Condensate and Air Return in Steam Heating Systems, by F. C. Houghten and J. L. Blackshaw, A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 199.

valve. The pressure in the supply main built up at a rate determined by the opening of the valve, rate of condensation and air elimination until finally it became stabilized under control of the reducing valve at about 3 lb per sq in. gage. For the test in Fig. 6 the supply line pressure was about 2 lb per sq in. gage at the time of maximum condensation. In this test the ratio of air to condensate at the peak is difficult to determine, due to the pump not running during the 3-min period previous to the peak. This places a larger removal load into the next 3-min period and makes the air and condensate loads at 33 min high. Using the values as plotted, this ratio would be 0.14 cu ft of air per pound of condensate. If the air removal values are averaged for the two periods ending at 30 and 33 min, this ratio would be 0.07. Should the peak at 42 min be considered the true condensation peak, the ratio would be 0.026, which is checked by the values of 0.033 from Fig. 7 and 0.028 from Fig. 8. In these latter tests there was no 3-min period in which the pump did not run. The supply main pressure was about 2 lb per sq in. gage at the time of the condensation peak for the test of Fig. 7, and only slightly above atmospheric for that of Fig. 8.

SUMMARY

During normal operation of a vacuum heating system having a connected capacity of 38,782 sq ft of EDR with about 30,500 sq ft active, the condensate load was 2614 lb per hour and the air removal load 87.6 cu ft per hour, which is a ratio of 0.034 cu ft of air per pound of condensate.

Under peak load conditions obtained by turning steam into the cold system, the ratio of air to condensate at the time of maximum condensate return was found to be 0.033 on a test with 30 min opening of the valve and 0.028 with 4 min opening of the valve. The difference in these values is not of any significance, and the ratio obtained depends to some extent on the conditions of operation and of testing.

In each peak load test the maximum air removal rate preceded the maximum condensate rate so that the two loads did not coincide. With the slower opening of the steam valve, the air peak occurred at 15 min and the condensate peak at 33 to 36 min. The corresponding values for rapid opening of the steam valve were 6 and 21 min.

The maximum condensate rate was slightly over 2.5 times the rate after normal conditions had been established for slow opening of the valve and about 3 times the normal rate for rapid opening of the valve. The corresponding ratios for air removal were about 21 and 28.

The vacuum pump used in these tests had a rated capacity of 90 gpm and 34 cfm at 10 in. vacuum and a capacity to serve 65,000 sq ft of radiation. During 8 days of normal operation with an outside temperature of 26.6 F, 5.8 per cent of the condensate capacity and 4.3 per cent of the air capacity were used. During peak load tests on a cold system the pump ran about 30 per cent of the first 2 hours with slow opening of the steam supply valve with an outside temperature of 42.5 F and 32.5 per cent of the time with an outside temperature of 32 F. With rapid opening of the valve and an outside temperature of 49.5 F, the pump ran 14 per cent of the time.

With slow opening of the valve and an outside temperature of 32 F, the maximum condensate measured in 3 min was at the rate of 12,000 lb per hour,

which is about 27 per cent of the pump capacity on condensate. The air removal peak was 7 per cent over the rated air handling capacity. With rapid opening of the valve and an outside temperature of 49.5 F, the maximum condensate rate in 3 min was 6800 lb per hour, which is about 15 per cent of the condensate handling capacity. The air removal peak in this test was 30 per cent over the rated air capacity due to the building up of a positive pressure in the return piping. At no time during the peak load tests did the pump run under float control to remove condensate as long as 1 min in any 3-min period.

Had such tests been made in the coldest weather, the above condensate peaks would have been greatly exceeded. In these tests slow opening of the valve corresponded to a 30-min period and rapid to a 4-min period from closed to fully open. There seems little reason for a period as short as the latter in the operation of a large heating system. The air peaks obtained depend upon the rate of entrance of steam to the system and the capacity of the pump to remove the air. A limited capacity on starting from cold conditions is perhaps desirable in order to reduce the strains due to heating up of piping and radiation.

ACKNOWLEDGMENTS

The results presented in this paper were obtained in the Department of Mechanical Engineering of which G. L. Larson is Chairman and Professor of Steam and Gas Engineering. Acknowledgment is also due to J. A. Dickie for the material collected in 1934-35 for his Master's Thesis, with G. Androne and R. A. Wilson assisting. The following assisted on the project during 1935-36: G. R. Anderson, A. J. Beyer, M. B. Conrad, L. W. Griffith, B. M. Kluge, C. L. Sawyer, R. A. Sharp and E. L. Wefel.

DISCUSSION

H. E. ADAMS: Professor Nelson's paper is an interesting collection of data pertaining to this particular subject. We should have more papers on this subject as the Society's fund of information on the rate of the return of air and condensate removal from vacuum heating systems is very meager.

I believe it should be emphasized to the members of the Society that the results of this paper pertain to an isolated case and that it would be dangerous to make a general application of the values observed. This is because of the rather peculiar nature of the heating system and the condition under which it is operating.

You will note that the total radiation given as some 38,000 sq ft had only some 11,000 sq ft of direct radiation, which is approximately a ratio of ⅓ direct to ⅔ indirect radiation. Furthermore, the indirect radiation was operated with full steam pressure on the radiation up to the trap at all times with temperature control effected by the starting and stopping of the attached fan, whereas, the steam for the direct radiation was supplied intermittently to the radiation by means of thermostatic control of the radiator inlet valves.

You will note also that the average condensation rate was 0.0857 lb per square foot of radiation per hour as compared to a generally accepted low average figure of 0.250 lb per square foot of radiation per hour. In other words, condensation rate

was ⅓ the average rate experienced. This would give the impression that the building was over radiated, even when we give consideration to the fact that the tests were not run at the designed outside temperature conditions.

It would seem that the radiation load of this building was more nearly in the neighborhood of from 20,000 to 25,000 sq ft.

I would call attention also to the ratio of cubic feet of air per pound of condensate although this method of expressing air capacity is not usual practice. Professor Nelson obtained a value of 0.034 cu ft of air per pound of condensate, which compares to usually accepted practice on direct radiation of a ratio of approximately 0.1 cu ft per pound of condensate. In other words, his ratio of air to condensate was approximately ⅓ of that normally experienced on direct radiation. I would again call attention to the fact that ⅓ of the total radiation on this building was direct radiation, and would point out the possible significance of this coincidence.

I would furthermore call attention to the fact that Professor Nelson has taken the volume of air at atmosphere in each case and in making his comparison in reference to the vacuum pump, he has taken the rated figure of the vacuum pump as 34 cfm at atmospheric pressure, whereas, the vacuum pump rating is at 10 in. Hg vacuum. His figures when compared to the percentage of the vacuum pump rating should therefore be corrected in the ratio of the absolute pressures—$\frac{30}{20}$ or a correction factor of 150 per cent.

The method adopted for measuring the air capacity is a relatively simple one and is all right for a rough check. It is not, however, strictly an accurate method of measuring the displacement of the pump unless tied in more definitely with vacuum and temperature conditions.

The temperature of the returns should also be stated for better defining of conditions as this has a large influence on the displacement required to remove the necessary air and vapor. The temperature, as I understand it, in this case was around 140 F—a rather low temperature due no doubt to the long strung out return lines. This I would point out is a favorable factor toward lowered air capacity.

PROF. D. W. NELSON: It is interesting to note the comments of Mr. Adams, a specialist in the field of vacuum pump manufacture. We had not expected these results to apply to all types of vacuum systems, but presented them to add to the meagre knowledge of the subject. Strictly speaking, the results apply only to the Mechanical Engineering Building, and more generally to a type having about ⅓ cast-iron direct, and ⅔ in the form of blast coils, unit heaters and unit ventilators. Such proportions, however, would seem to exist in a large number of school buildings and in industrial buildings that combine laboratories, offices and shops. The results would not apply to an office building with cast-iron radiation predominating.

The data in Table 1 do indicate 0.0857 lb condensation per square foot for a normal equivalent radiation of 30,500 square feet at an average temperature of 27 F. This indicates a condensation rate of 0.17 lb per square foot at the design temperature of —15 F. The heat losses were calculated and the radiation was chosen by the practice established in the Society's GUIDE. The allowance made for infiltration would tend to lower the condensation rate since infiltration is figured on all sides of the building but obviously cannot act on all sides simultaneously. Further, the fan system is run only during the day and unit ventilators recirculate at night. Direct radiation is carried at a lowered night temperature. These variations account for the reduction to 0.17 lb condensation per square foot from the normal laboratory figure of 0.25 lb per square foot for radiation, rather than the simple statement that the building was over radiated. All of the 30,500 square feet of radiation figured as normal is required

under design conditions, and it is in the interest of good operating economy to reduce heating and ventilating costs during periods of non-occupancy. The equivalent direct radiation used throughout the 24 hours would apear to be 0.17/0.25 x 30,500 or 20,700 sq ft.

The method of expressing air capacity in the ratio to condensate was used to give direct comparison with the previous paper of the Society's laboratory staff on a similar study. That paper indicated 0.05 cu ft per pound condensate as compared to the figure of 0.034 cu ft in the present study for the same type of operation. This indicates a considerably lower air load than the 0.1 cu ft quoted by Mr. Adams for direct radiation. The tests indicate that the latter figure should be safe for pump designers to follow as it appears to have a considerable margin of safety. It should be borne in mind that the 0.034 figure was obtained on a system containing among more normal elements 4430 sq ft of cast-iron wall radiation in units having as many as 26 sections each, and 29 ft long. Fig. 3 shows that on one day during the early morning hours of lowered temperatures, the pump was required to run continuously due to air leaks into a cold system stated by Mr. Adams as being too light to be representative.

The rated air capacity of the vacuum pump of 34 cfm from 10-in. vacuum was mistakenly taken as measured at standard conditions. Mr. Adams is therefore correct in calling attention to a correction factor of 150 per cent to apply to the air removal factors in the last column of Table 1. Instead of ranging from 0.04 to 0.05 these would be from 0.06 to 0.07. The average use of the air capacity at a 26.6 F temperature would be 6.44 per cent instead of 4.33 per cent. The pump has more than ample capacity for all average demands.

The method adopted for measuring air capacity, stated by Mr. Adams as being relatively simple and all right for a rough check, is considered to be within the accuracy called for in the tests. Pressures, temperatures and orifice coefficients were used in these calculations. The set up shown in Fig. 2 was developed for this field study after performing many laboratory tests on two vacuum pump test arrangements. A very large number of man hours went into the assembling and the using of this equipment to get the results reported in this paper. As previously mentioned, five observers were used on peak load tests taking 1-min. readings of all variables.

In conclusion, we wish to say that there appears to be no need of warming up a system as rapidly as was done in the test reported in Fig. 8, and probably not even as fast as in Figs. 6 and 7. There is some virtue in the capacity of the pump being limited so that the cold system will not be freed of air too quickly since slow removal allows more gradual warming up to temperature with less expansion strain.

No. 1059

SEMI-ANNUAL MEETING, 1937

ONE of the largest Semi-Annual Meetings of the Society was held at the New Ocean House, Swampscott, Mass., during the week of June 24-26, where members and guests totalling 380 were in attendance from all parts of the United States and Canada to enjoy the entertainment program provided by the Massachusetts Chapter and the technical discussions of a group of interesting subjects.

Two Council meetings were held during the sessions at which time it was announced that favorable action had been taken upon the application for the Golden Gate Chapter charter in San Francisco, Calif., and that the Semi-Annual Meeting 1938 was to be held June 20-22 at The Homestead, Hot Springs, Va. In addition to the general Research Committee luncheon and meeting on Thursday, several technical advisory groups met to consider the instigation of new research projects. Members of the Guide Publication Committee were in full attendance at their dinner session held Wednesday evening, and the Nominating Committee convened to select Society officers and Council members for the year 1938.

Pres. D. S. Boyden, Boston, Mass., called the first session of the Semi-Annual Meeting 1937 to order Thursday morning at the New Ocean House, Swampscott, Mass. Prof. James Holt, president of Massachusetts Chapter, welcomed the members, guests and ladies to New England and expressed appreciation for the large number in attendance.

A brief response was made by President Boyden, who then announced the first technical paper, The Adaptability of Precooling Coils to Air Washer Systems, by John Everetts, Jr., which was presented by W. L. Fleisher, New York, N. Y. S. S. Sanford, Detroit, Mich., presented the paper, How Much Power Will an Air Conditioning System Use, by A. D. Marston. Cooling Load Analysis of a Bank Building, by John James, New York, N. Y., was presented by the author. The Study of Summer Cooling in the Research Residence Using Water at Temperatures of 52 F and 46 F, by A. P. Kratz, S. Konzo and E. L. Broderick was presented by S. Konzo.

L. A. Harding, Buffalo, N. Y., chairman of the Joint A.S.H.V.E.—*A.S.R.E.* Committee on National Standards for Air Conditioning Applications, read the Proposed Code of Minimum Requirements for Comfort Air Conditioning which was ordered printed and distributed to the members for letter ballot

The second session was called to order on Friday, June 25, by President Boyden, who announced the first paper, Calculated Over-all Coefficients for Walls with Air Space Insulation, by F. B. Rowley.

President Boyden introduced Prof. G. B. Wilkes, Cambridge, Mass., who

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presented the paper, Radiation and Convection Across Air Spaces in Frame Construction, by G. B. Wilkes and C. M. F. Peterson.

Prof. G. L. Tuve, Cleveland, Ohio, presented the paper, Performance of Fin-Tube Units for Air Cooling and Dehumidifying, by G. L. Tuve and C. A. McKeeman.

The Cooling and Heating Rates of a Room with Different Types of Steam Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick, was presented by Professor Fahnestock, Urbana, Ill.

The third and final session of the Semi-Annual Meeting was opened on Saturday morning by President Boyden. H. C. Murphy, Louisville, Ky., was introduced and read the paper, An Alternate Method of Comparing the Dust Arrestance of Air Cleaning Devices, by Arthur Nutting.

President Boyden introduced Prof. C. P. Yaglou, Boston, Mass., who gave the paper, Ventilation Requirements (Part 2), by C. P. Yaglou and W. N. Witheridge.

Prof. R. R. Seeber, Houghton, Mich., presented the paper, Corrosion Studies in Steam Heating Systems, by R. R. Seeber, and Margaret R. Holley. President Boyden introduced J. H. Walker, Detroit, Mich., who outlined the history of the present research investigation and told of the financial cooperation the Society had received from several interested organizations.

The final paper of the meeting, Heating Requirements of an Office Building as Influenced by the Stack Effect, by F. C. Houghten and Carl Gutberlet, was given by F. C. Houghten, Pittsburgh, Pa.

Amendment to Constitution and By-Laws

Secy. A. V. Hutchinson read the Amendment to the By-Laws of the Society as approved by the Council.

Article B-II, Section 5. Any person who has been a member of the Society for fifteen (15) years or more upon reaching the age of seventy (70) shall have his dues remitted for the current year and for ensuing years, without surrendering any of the privileges of membership as long as he lives.

To Be Amended as Follows: Any Member or Associate Member who has paid annual dues as a Member or Associate Member for thirty (30) years and who has reached the age of sixty-five (65) shall be eligible for Life Membership.

It was moved by F. C. McIntosh, Pittsburgh, Pa., and seconded by Albert Buenger, Dayton, Ohio, that the amendment be adopted. The motion was approved unanimously.

President Boyden then called W. T. Jones, Boston, Mass., to the rostrum and requested him to install the first vice-president, who was on a round-the-world trip at the time of the last Annual Meeting. Mr. Jones called upon Mr. Carrier to act as marshal, and E. H. Gurney, Toronto, Can., was escorted to the platform where he was formally installed as first vice-president of the Society.

Resolutions

The following resolutions were presented by N. D. Adams, Rochester, Minn., and unanimously adopted:

Before closing this most pleasant and instructive meeting, we, the members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, assembled in Swamp-

scott, Mass., for the Semi-Annual Meeting 1937, wish to thank the many who are responsible for making it an outstanding success. Therefore,

Be It Resolved, That we feel deeply grateful to the weatherman, who has given us such perfect days and moonlight nights,

That we are indebted to the Massachusetts Chapter, who, under the able leadership of their general chairman, W. T. Jones, and his Committee chairman, have worked so faithfully and efficiently,

That we appreciate the messages brought to us by the very able speakers at our technical sessions and at our banquet, to stimulate us through the coming months,

That we thoroughly enjoyed the scenic trips brought to us through pictures taken by our president, D. S. Boyden, and L. C. Soule, who have proven themselves to be real artists in choosing the worthwhile sights along the way,

That we are appreciative of the excellent service and courtesies extended by the management of the New Ocean House,

That we thank the press for its generous support in giving space and publicity to the activities and the worthwhile papers which have been presented at our sessions,

That we express our pleasure to the Canadian members for their thoughtful courtesy in presenting their banner, which will float beside our Stars and Stripes at all future meetings and thereby further cement the neighborly feeling between us and emphasize the international character of the Society, and

Be It Further Resolved, That to all who have contributed toward making this meeting such a happy one we want to say that we are glad we have come and will long remember this outstanding meeting.

PROGRAM SEMI-ANNUAL MEETING, 1937

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW OCEAN HOUSE, SWAMPSCOTT, MASS.

(Daylight Saving Time)

JUNE 23-26, 1937

Wednesday, June 23

1:00 P.M. Registration.
2:30 P.M. Council Meeting.
6:30 P.M. Guide Publication Committee.
8:30 P.M. President's Reception, Music and Dancing.

Thursday, June 24

9:00 A.M. Registration.
9:00 A.M. Nominating Committee.
9:30 A.M. *Technical Papers:*
The Adaptability of Pre-Cooling Coils to Air Washer Systems, by John Everetts, Jr.
How Much Power Will An Air Conditioning System Use? by A. D. Marston.
Study of Summer Cooling in the Research Residence Using Water at Temperatures of 52 F and 46 F, by A. P. Kratz, S. Konzo and E. L. Broderick.
Cooling Load Analysis of a Bank Building, by John James.
Report of A.S.H.V.E.-*A.S.R.E.* Committee on National Standards for Air Conditioning Applications, by L. A. Harding, Chairman.
12:30 P.M. Committee on Research.
1:30 P.M. Boat Trip to Gloucester.
2:00 P.M. Golf Tournament—Tedesco Country Club (18-hole, Kickers Handicap).
8:30 P.M. Informal Party, Concert and Dancing.

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Friday, June 25

9:30 A.M. *Technical Papers:*
Ventilation Requirements, Part II, by C. P. Yaglou and W. N. Witheridge.
Calculated Over-all Coefficients for Walls with Air Space Insulation, by F. B. Rowley.
Radiation and Convection Across Air Spaces in Frame Construction, by G. B. Wilkes and M. F. Peterson.
Performance of Fin-Tube Units for Air Cooling and Dehumidifying, by G. L. Tuve and C. A. McKeeman.
The Cooling and Heating Rates of a Room with Different Types of Steam Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick.

10:00 A.M. Bus leaves for Boston—Ladies to visit Harvard Museum and Glass Flower Exhibit.

2:00 P.M. Golf Tournament, Tedesco County Club (18-hole, Medal Play—Research and Chapter Cups).

2:30 P.M. Ladies' Bridge and Tea—Tennis, Archery and other sports.

7:30 P.M. Banquet (New England Shore Dinner) and Dance—(Quartette and Speaker)—(Presentation of Past President's Memory Book).

Saturday, June 26

9:30 A.M. *Technical Papers:*
An Alternate Method of Comparing the Dust Arrestance of Air Cleaning Devices, by Arthur Nutting.
Corrosion Studies in Steam Heating Systems, by R. R. Seeber and Margaret R. Holley.
Heating Requirements of an Office Building as Influenced by the Stack Effect by F. C. Houghten and Carl Gutberlet.

10:00 A.M. Ladies' visit to North Shore Gardens.

COMMITTEE ON ARRANGEMENTS

W. T. JONES, *General Chairman.*

Registration and Reception: A. S. KELLOGG, *Chairman;* R. A. SPENCE, C. R. SWANEY and H. B. WIEGNER.

Entertainment and Ladies: JAMES HOLT, *Chairman;* C. T. FLINT, H. C. MOORE and JOHN TURNER.

Transportation: T. F. MCCOY, *Chairman;* G. B. GERRISH and W. A. MCPHERSON.

Finance: W. E. BARNES, *Chairman;* J. W. BRINTON and EDGAR SHAW.

Golf: J. F. TUTTLE, *Chairman;* R. S. FRANKLIN and A. F. KARLSON.

Publicity: R. M. NEE, *Chairman;* E. T. KEEFE and H. F. WHITTEN.

No. 1060

THE ADAPTABILITY OF PRECOOLING COILS TO AIR WASHER SYSTEMS

By JOHN EVERETTS, JR.* (*MEMBER*), NEW YORK, N. Y.

THE use of cooling coils in an air conditioning system has long been recognized as having certain definite advantages over a system employing only an air washer. This is particularly true of comfort conditioning where the exact simultaneous control of temperature and humidity is not as important as it is in an industrial installation.

A system using a saturation type air washer and reheat, however, can be designed to meet any exacting requirements, because the absolute moisture content, which is a function of the dew-point, is obtained by saturating the air at a predetermined temperature and the dry-bulb is controlled by reheat. It is the aim of this paper to show the advantage of using both a cooling coil and an air washer in the same system.

The psychrometric characteristics of a vapor-air mixture are such that the moisture load and sensible load must be considered as separate entities in any heat transfer problem where these two factors vary to any degree. In an air washer, the heat transfer reaction necessary to fix the absolute moisture content is to first adiabatically saturate the air by intimate contact between the vapor-air mixture and the atomized spray, and then a sensible heat transfer between the saturated mixture and the spray water. The heat transfer efficiency is a function of (1) the total heat to be removed, (2) the intimacy of contact between the vapor-air mixture and the spray water, (3) the length of time of contact, and (4) the surface area of the spray water to which the vapor-air mixture can dissipate its heat. The total heat to be removed is, of course, a function of the difference between the entering and leaving wet-bulb temperatures. The intimacy of contact is dependent upon the degree of atomization of the spray water, and the length of time of contact is a function of the length of the washer and the air velocity through the washer. The contact surface area is a function of the atomization and quantity of water sprayed.

A good commercial air washer to do the dehumidifying, cooling and saturating in an industrial installation will require approximately 12 to 15 gpm of spray water per 1,000 cfm with a temperature rise in the water of from

* Air and Refrigeration Corporation.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1937, by W. L. Fleisher.

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5 to 6 F. Spray nozzle pressures of from 18 to 30 lb are necessary to sufficiently atomize the water for this duty. Assuming a saturation efficiency of 90 per cent, the leaving water temperature for 10 F entering wet-bulb depression would be approximately one degree lower than the leaving wet-bulb of the air, and the leaving dry-bulb one degree higher than the leaving wet-bulb. This means the entering water must be 6 to 7 F lower than the required leaving wet-bulb. As the air leaving the washer is within one degree of saturation, the dew-point will be within approximately one-half a degree of the leaving water temperature. It must be kept in mind, however, that if the total heat removal is such that a rise in the spray water temperature of more than 6 F is required, then a two-stage or multi-stage washer is necessary in order to keep the terminal temperatures of air and water within the prescribed limits. As the air velocity, water quantity, outlet water temperature, and pressure on the nozzles are kept constant, keeping the leaving saturation temperature constant for a decreased load will result in a rise in the inlet water temperature to balance this load.

A cooling coil, however, has entirely different characteristics from an air washer. The transfer of heat from the vapor-air mixture to the water is made through an intermediate surface of fixed area. Assuming the air velocity to be constant, the leaving condition of the air will vary with a change in either and both the entering dry-bulb and wet-bulb, change in water velocity, and change in total heat removal which affects the temperature rise of the water in the coil. The heat removed from the air by the water in the coil is total heat, the same as in the case of a washer so long as the entering dew-point is above the required leaving dew-point, and the coil may be designed to give a saturated leaving air condition under certain entering conditions. However, if the entering wet-bulb is higher than the required leaving wet-bulb, as shown in Fig. 1, thus requiring cooling effect, and the entering dew-point is lower than the required leaving dew-point, the heat removal is all sensible heat affecting only the dry-bulb temperature and resulting in an indeterminate delivered dew-point. However, as the total heat removal is a function of the variables affecting the heat transfer, then a coil may be designed to maintain a constant leaving wet-bulb temperature so long as this temperature is lower than the entering wet-bulb temperature. With a fixed coil surface area, the external variables affecting the heat transfer rate are (1) air velocity, (2) dry-bulb temperatures, and (3) dew-point. An increase in air velocity increases the heat transfer because it tends to reduce the film layer on the coil surface thus reducing the film resistance. An increase in dry-bulb temperature increases the heat transfer because of the increased mean effective temperature difference (M. E. D.) between the air on the outside and the water on the inside. An increase in dew-point may or may not increase the heat transfer rate. If the dew-point is above the coil surface temperature, condensation will take place and wet the surface of coils which would increase the rate of heat transfer; however, if a finned coil is used with the fins so close together that the water builds up between them, the effective surface area of the coil is reduced, thus decreasing the heat transferred.

The internal variables are (1) water velocity which governs the quantity, and (2) water temperature. The water velocity affects the heat transfer by changing the thickness of the film layer on the inside of the tube, and the

quantity of water affects the M. E. D. because of the quantity of heat absorbed which governs the temperature rise. The temperature governs not

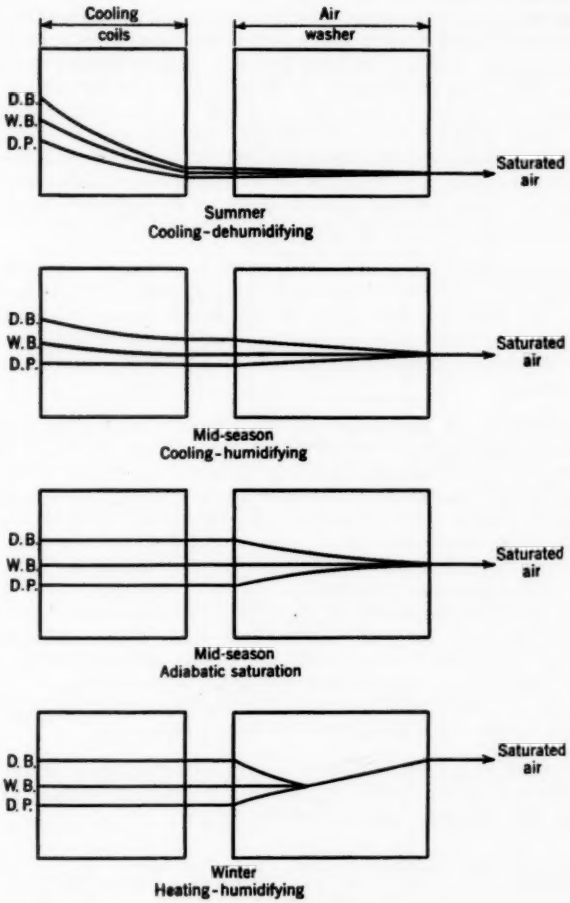

FIG. 1. AIR TEMPERATURE VARIATIONS THROUGH PRECOOLING COILS AND AIR WASHERS FOR DIFFERENT SEASONS

only the M. E. D. but the amount of wetted surface due to condensation for a given dew-point of the entering air.

In spite of the many variables encountered in the proper design of a cooling

system the coils have many advantages over a spray type dehumidifier. The heat exchange is counter-flow in a cooling coil, whereas in a washer it is cross-flow. That is, the water temperature leaving the cooling coil may be higher than the air temperature leaving the coil as compared to an air washer where the leaving water must be lower than the leaving air. This directly affects the suction pressures of a refrigerating system and also makes it possible to use well water in many cases. The heat transfer characteristics are such that a temperature rise of the chilled water of 10 to 15 F may be obtained in a cooling coil, whereas in a washer a rise of only 5 to 6 F may be obtained resulting in a circulated water quantity of from ½ to ⅓ of that required in a washer. The water friction through the coils is considerably less than the pressure required on air washer nozzles, thus resulting in additional pump power saving. One of the most important characteristics, however, is the fact that the air does not come in direct contact with the chilled water.

In certain industrial plants where the water is cooled in a central refrigerating system and distributed to local conditioning units, it might be disastrous if the odors or flavors picked up by the air washer water in one department were allowed to circulate in an air washer in an entirely different department. The coils or closed systems are the only alternative to this frequently disturbing problem.

In the design of an air washer system with precooling coils, it is usually preferable to design the coils to carry the full cooling load. As the cooling coil surface varies with the M. E. D., and the water quantity varies with the total heat, the coil should be designed with the lowest dry-bulb temperature which may frequently occur coincident with the design wet-bulb. That is, if the general design conditions for air conditioning are 90 F dry-bulb and 75 F wet-bulb, then the coil should be designed for a condition of 75 F saturated air, which occurs almost as frequently as the 90-75 condition. The 15 F difference in the outside temperature may not necessarily reduce the total load in the conditioned spaces by more than a few per cent, because in industrial air conditioning, the product load, people and motors are so great a proportion of the total load that the transmission load is not the predominant factor. This is not always the case, however, and it is, therefore, good practice to check these values before considering the coil design conditions.

The following design conditions of a system, as well as those in Table 1, have been selected to illustrate some of the pertinent points which must be considered in designing a combination coil and washer system:

Outside Air —90 F Dry-Bulb—75 F Wet-Bulb
Return Air —70 F Dry-Bulb—50 per cent Relative Humidity
Delivered Air —46 F Saturated

Case I — 15 per cent Outside Air—85 per cent Return Air
Case II — 50 per cent Outside Air—50 per cent Return Air
Case III—100 per cent Outside Air— 0 per cent Return Air
Case IV—100 per cent Outside Air— 0 per cent Return Air
—Outside Wet-Bulb—46 F

In Case I where only 15 per cent outside air is used, there is no increase in coil surface with lowering of the outside dry-bulb temperature and with fixed

design wet-bulb. The reason for this is that the net change of dry-bulb of the entering mixture is very small, although it is sufficient to reduce the sensible heat load to compensate for the lower K factor, thus giving approximately the same surface area.

In Case II the drop in outside dry-bulb temperature is more apparent in the mixture and requires approximately 3 per cent more surface at the 75 F saturated condition of outside air than at the condition of 90 F dry- and 75 F wet-bulb.

In Case III where all outside air is used, the increase in surface caused by the decrease in dry-bulb temperature is approximately 12 per cent or enough

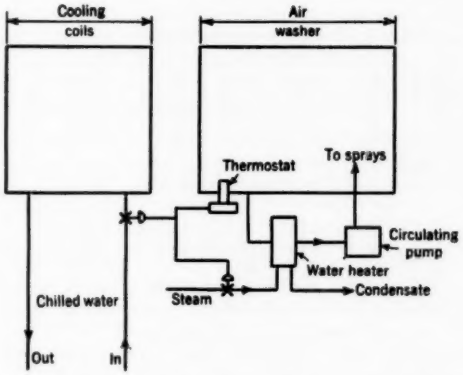

FIG. 2. AUTOMATIC CONTROL OF PRECOOLING COILS AND AIR WASHER

to seriously affect the proper operation of a system when a 75 F saturated condition occurs.

Case IV illustrates the point previously referred to in the paper that if the wet-bulb entering the coil is at or below the required delivered dew-point and the entering dry-bulb is above this point, there can be no control of the delivered dew-point. As it is necessary to reduce the entering dry-bulb, it is necessary to use refrigeration effect as indicated. If an air washer is used in conjunction with the coils, the air at conditions shown in Case IV could be adiabatically saturated at the entering wet-bulb, thus eliminating the necessity of refrigerating effect and definitely controlling the delivered dew-point at the same time. The annual saving in operating costs of the refrigerating equipment is usually greater than the added cost of a saturation type washer.

The power required to operate the saturation type washer in conjunction with the coils is only 2 to 5 per cent greater than the total power requirements of the normal system taking into consideration the saving in power of the smaller chilled water pump due to the smaller water quantity and smaller total

head. The fan power is increased by the resistance of coils on the washer, and may or may not make a considerable difference in the total power, depending upon the type and design of the air handling system.

In many industrial problems, it is advantageous to remove from the air the

TABLE 1—DESIGN CONDITIONS FOR COMBINATION COIL AND WASHER SYSTEM

Outside Air Temperature in Deg F	Air Temperature Entering Coil in Deg F			Air Temperature Leaving Coil in Deg F			Total Heat Removed Btu/Hour	k Factor Btu/Hour/Sq Ft/Deg F	Surface Sq Ft
	D.B.	W.B.	D.P.	D.B.	W.B.	D.P.			
	Case I—15 Per Cent Outside Air								
90—75	72.5	61.3	54.2	47.0	46.0	45.0	3060	153	970
75 Sat.	70.8	61.3	55.5	47.0	46.0	45.0	3060	149	968
70 Sat.	70.0	60.5	54.7	47.0	46.0	45.0	2870	152	962
60 Sat.	68.0	58.7	52.3	47.0	46.0	45.0	2460	153	825
	Case II—50 Per Cent Outside Air								
90—75	80.0	67.5	61.3	47.0	46.0	45.0	4570	146	1190
75 Sat.	72.5	67.5	65.1	47.0	46.0	45.0	4570	132	1225
70 Sat.	70.0	64.7	61.1	47.0	46.0	45.0	3840	135	1080
60 Sat.	65.0	59.0	55.0	47.0	46.0	45.0	2580	142	930
	Case III—100 Per Cent Outside Air								
90—75	90.0	75.0	69.0	47.0	46.0	45.0	6700	139	1465
75 Sat.	75 Sat.			47.0	46.0	45.0	6700	115	1640
70 Sat.	70 Sat.			47.0	46.0	45.0	5400	117	1390
60 Sat.	60 Sat.			47.0	46.0	45.0	2740	123	980
50 Sat.	50 Sat.			47.0	46.0	45.0	714	133	378
	Case IV—100 Per Cent Outside Air								
60—46	60.0	46.0	30.0	47.0	40.0	30.0	1065	170	615
55—46	55.0	46.0	32.0	47.0	42.0	37.0	658	170	450
50—46	50.0	46.0	42.5	47.0	44.5	42.5	246	170	212
46 Sat.	46 Sat.			46.0	46.0	46.0	.....	.....	.....

soluble gases, and other impurities which filters and cooling coils cannot take out, but which can be removed by an air washer.

The automatic controls necessary for the proper summer and winter operation of a combination of precooling coils and air washer are greatly simplified, as indicated in Fig. 2, over the conventional system of controls required for this duty. The thermostat, located either in the washer tank, washer outlet,

or pump suction, controls the quantity of chilled water to the cooling coils through the throttling valve, or the quantity of steam to either the water heater or an air pre-heater to control the wet-bulb of the air entering the washer. It must be kept in mind, however, that the washer must be of the saturation type in order that this control may adequately fix the dew-point of the air leaving the washer.

It is much simpler in this type of system to drain the washer, hose it down with fresh water and refill it, than it is to completely change the water in the chilled water circuit as would be necessary if the chilled water were not in a closed system.

The winter operation of such a system is the same as a conventional air washer system.

The reheat requirements in the summer are also the same as in a conventional air washer system in which either low pressure steam or a by-pass may be used. If a by-pass is used, it is preferable to use a washer of full fan capacity to take advantage of evaporative cooling and saving in refrigerating operating costs when the outside wet-bulb is suitable.

With the refined development of cooling coils and the established dependability of the air washer, the combination of the two properly designed brings the problem of industrial air conditioning and comfort cooling nearer to the criterion of perfection.

SUMMARY

1. The combination of precooling coils and an air washer gives the accumulated advantages of each and none of the disadvantages to a system where exact year-'round control is required.
2. The use of coils alone will not give a control of the delivered dew-point unless the entering dew-point is higher.
3. The use of an air washer alone is limited in cooling capacity and under certain conditions cannot carry the load unless multi-staging is used.
4. The combination of cooling coils and air washer eliminates these two major objections, and gives positive year-'round control of air conditions.

DISCUSSION

DR. B. M. WOODS: Aside from the merits of the combination proposed by Mr. Everetts, for which he makes a good case, I desire to comment on the heat transfer rate for cooling coils. The question of film resistance, whether of the film between the liquid and the coil surface on the one side or the air film on the other, is one meriting careful examination for design; whichever has the lower transfer rate becomes the bottleneck, so to speak, of the equipment. For example, recent tests of unit gas heaters, in which finned tubes serve as flues for the products of combustion and the exterior finned surfaces serve to warm air circulated over them, have shown that the real need for a finned surface is on the inside where the flue gases are. Increase of outside fin area can do practically nothing for this equipment.

The author wisely alludes to a second point; namely, that increased velocity of flow increases heat transfer. It is important to reach turbulent, instead of laminar, flow. This is usually easy to do.

The principles of heat transfer are not difficult to state, but they are easy to

overlook in design. The example of the Velox boiler, now coming into considerable use in Europe, is of interest. This steam generator has provided an increase in heat transfer rate of about 1000 per cent by better use of heat transfer knowledge; the most important single development being positive circulation of boiler wastes at sufficient velocities to give turbulent flow. Mr. Everetts is to be congratulated on his method of analysis, with its emphasis on heat flow principles, as well as for his timely discussion of promising equipment combinations.

No. 1061

HOW MUCH POWER WILL AN AIR CONDITIONING SYSTEM USE?

By ANSON D. MARSTON * (*MEMBER*), KANSAS CITY, MO.

"HOW much will it cost to operate?" Whenever the installation of air conditioning is under consideration, this question is sure to arise. This is a logical question—one for which the profession should have a reasonable answer, but so far no satisfactory method has been developed for answering it.

For a number of years, the heating industry has had a satisfactory basis for estimating heating costs and for checking the operation of heating plants. By using degree-days below 65 F, excellent results have been obtained, with benefit to everybody concerned. This paper will outline the results of an investigation undertaken in an effort to arrive at some similar basis for use in estimating the energy consumption of air conditioning plants. As will be shown, there was found a strong indication that the *degree-days above* 65 *F* furnish an acceptable basis for such estimates.

One of the reasons why little information has been available on this subject is that comparatively few air conditioning plants were in operation prior to 1934. In view of the fact that accurate consumption records were kept for relatively few of the plants that were in operation, it is easy to understand why no general effort has been made to resolve the data available to some common denominator.

Theoretically it would seem improbable that the relationship between commonly furnished weather data and energy consumption would be close enough to be of value. The calculations necessary in determining the cooling load must take into consideration a number of items which are not included in heating problems and which are largely independent of weather conditions. Frequently the removal of heat and moisture, due to the presence of human beings, and of heat from special equipment such as lights and appliances, is a major part of the work to be done by a cooling plant. Obviously this must be done regardless of the weather. Sun effect is another important item whose magnitude is largely independent of temperature and humidity readings.

It must be remembered, however, that in a large majority of the smaller sized air conditioning installations, operation in practice is based upon temperature,

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* Air Conditioning Division, Kansas City Power & Light Co.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1937, by S. S. Sanford.

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without much regard to humidity. Few installations having a capacity of 20 tons or less are provided with any form of humidity control. In a large majority of cases, these installations are operated by people who watch the thermometer and who neither understand nor watch the relative humidity. Consequently, it appeared when these investigations were begun that temperature relationships were worth investigating.

Beginning in 1933, the Kansas City Power & Light Co. offered to its customers an optional rate for power used by strictly air conditioning equipment. This rate involves the use of a separate meter to measure the power used by such equipment. Furthermore this is an optional rate: wherever any other

TABLE 1—ELECTRICAL ENERGY CONSUMPTION OF AIR CONDITIONING CUSTOMERS

MONTH	SUMMARY							
	1934		1935			1936		
	AVERAGE OF KWHR PER HP		AVERAGE OF KWHR PER HP			AVERAGE OF KWHR PER HP		
	All Cust's	Comm'l Cust's	All Cust's	Res. Cust's	Comm'l Cust's	All Cust's	Res. Cust's	Comm'l Cust's
May	156	184	18	0	23	72	14	101
June	250	265	109	71	119	185	125	211
July	313	314	307	270	318	294	260	311
August	250	253	234	208	242	286	243	307
September	78	80	78	47	88	138	96	158
October	29	31	19	3	24	15	1	21
Totals	1076	1127	765	599	814	990	739	1109

rate is more economical, the consumption for cooling purposes is billed on the other rate.

Of special importance in this study is the fact that this rate calls for customers to be disconnected on October 15 of each year and reconnected on May 1 of the following year. Consequently, fans and other equipment, which are to be used during the winter, are usually served under one of the other rate schedules.

During the period from 1934 to 1936 inclusive, a total of 72 customers operated under this rate for periods long enough to be of value in this study. Many of these furnish records covering a part only of this period. However, enough information is available for each of these three years to serve as a basis for close study of the relationship between power consumption and weather. The records obtained from these customers are summarized in Table 1.

METHOD OF DETERMINING A DEGREE-DAY BASE

On pages 523 to 528 of THE A.S.H.V.E. GUIDE, 1937, will be found a discussion of the methods used in determining the base temperature for degree-

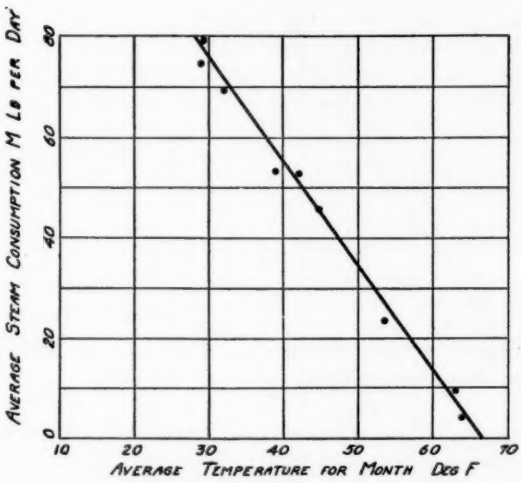

Fig. 1. Method of Determining Base Temperature for Degree-Day Calculations

day calculations for the heating season. Briefly the base temperature was selected originally by plotting average steam consumption against average monthly temperatures. Points plotted in this manner fall close to a straight line and indicate that consumption is directly proportional to the difference between 65 F and the average temperature. Fig. 1 is reproduced from page 527 in The Guide 1937, which illustrates the manner in which this was done.

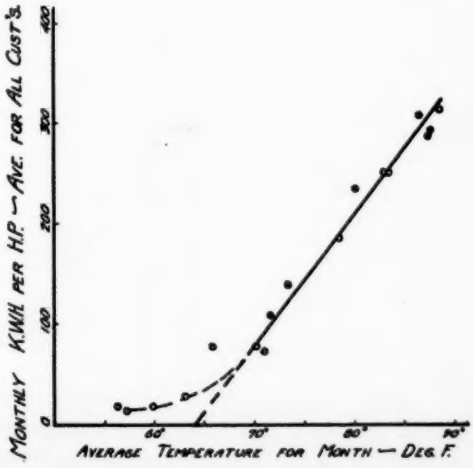

Fig. 2. Base Temperature for Degree-Day Calculations—Summer Cooling

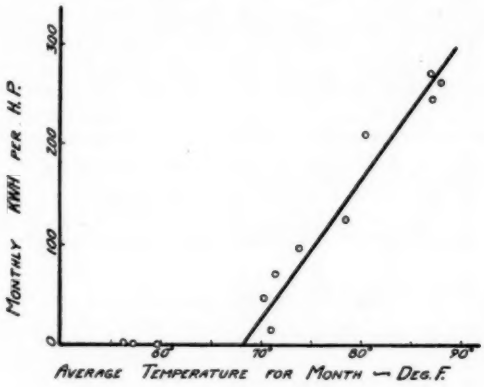

FIG. 3. EFFECT OF TEMPERATURE ON ENERGY USAGE AVERAGE FOR ALL RESIDENCES

In the 1932 Proceedings of the *National District Heating Association* there is reported a study of the steam consumption of 163 buildings. These were divided into 20 classes, depending upon usage, and the base temperature corresponding to zero steam consumption was determined for each class. These base temperatures vary from 61.2 F for auto sales and service buildings to 68.8 F for apartments. The average is 66.0 F for all buildings. However, in spite of this variation, it is found in practice to be satisfactory to use the 65 F base for all classes of buildings.

Similar investigations were made using the summer energy consumption data summarized in Table 1. Fig. 2 shows the relation existing between average

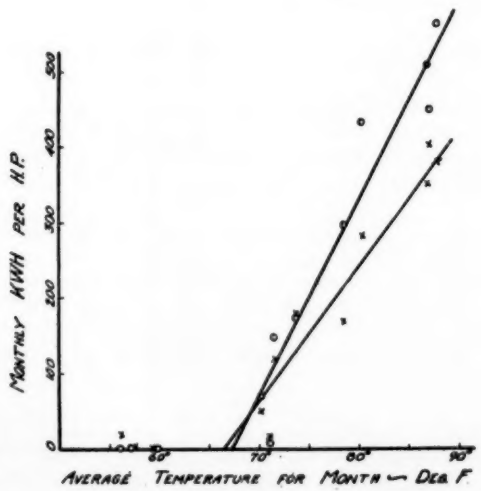

FIG. 4. EFFECT OF TEMPERATURE ON ENERGY USAGE—TWO TYPICAL RESIDENCES

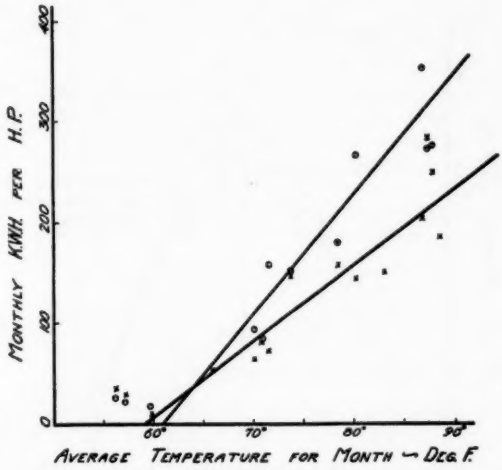

Fig. 5. Effect of Temperature on Energy Usage—Two Typical Offices

monthly temperatures and the average energy consumption for all classes of customers. A study of this chart will show that the consumption is substantially proportional to the difference between 65 F and the average monthly temperature. However, the figures for those months with average temperatures below 65 F deviate considerably from this relationship. This is to be expected, because many months with low average temperature contain periods of a week or more of hot weather with consequent use of cooling equipment. For example, September 1934 had an average temperature of 65.8 F, although it contained seven days with maximum temperatures of 85 F or higher.

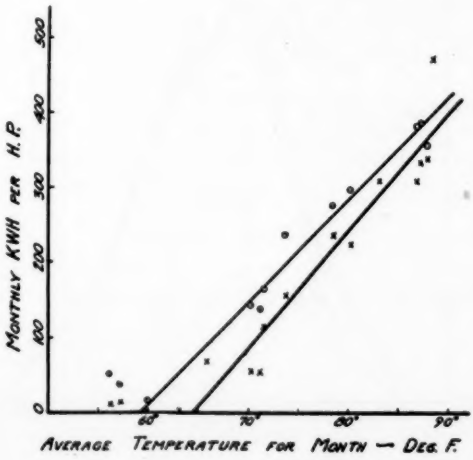

Fig. 6. Effect of Temperature on Energy Usage—Two Typical Restaurants

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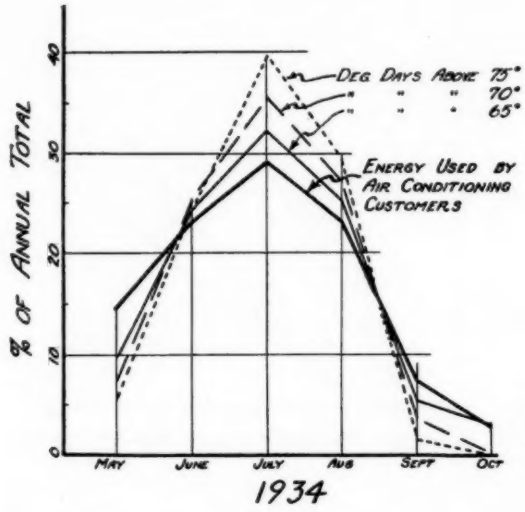

FIG. 7. MONTHLY VARIATION IN DEGREE-DAYS AND IN POWER USED BY AIR CONDITIONING CUSTOMERS, KANSAS CITY, MO.

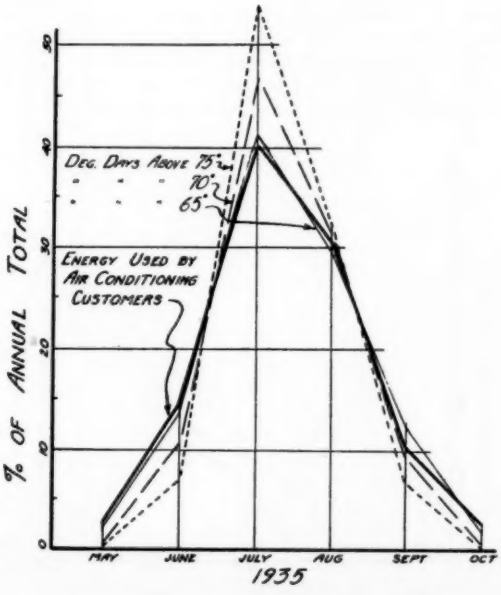

FIG. 8. MONTHLY VARIATION IN DEGREE-DAYS AND IN POWER USED BY AIR CONDITIONING PLANTS, KANSAS CITY, MO.

The same information for residential customers is shown in Fig. 3. It will be noted that the base temperature for residential customers appears to be 68 F.

As a further check on the relation between temperature and energy usage, Figs. 4, 5, and 6 were drawn. Fig. 4 shows the temperature relationship for two typical residences; Fig. 5 shows the data for two typical offices; and Fig. 6 shows the same information for two typical restaurants. Similar curves have been drawn for other customers and they indicate, in each case, a close relationship between average monthly temperature and energy consumption. Customers whose load is primarily determined by weather conditions—resi-

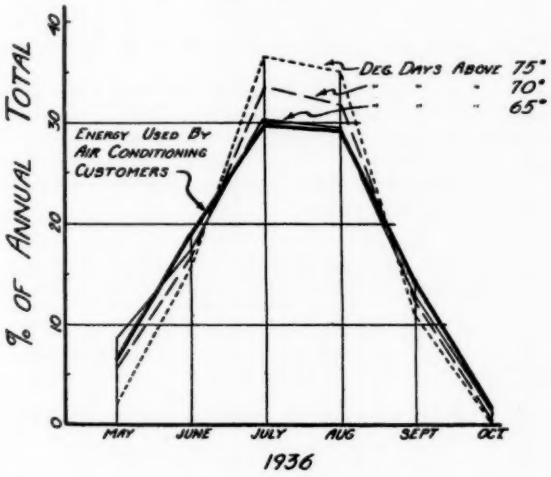

Fig. 9. Monthly Variation in Degree-Days and in Power Used by Air Conditioning Plants, Kansas City, Mo.

dences for example—have base temperatures above 65 F, while customers having a large proportion of their load independent of weather conditions generally have base temperatures below 65 F.

Additional Studies of Base Temperatures

When this study was started, it was expected that the base temperature would be between 70 F and 75 F. When an indicated base of 65 F was found, it was felt that additional studies should be undertaken as a verification. As a result of these studies, Figs. 7, 8, and 9 were drawn. In these figures the per cent of total annual energy consumption occurring during each month of the cooling season is plotted for each year. In the same figures is shown, for each month, the per cent of the total annual number of *degree-days above 65 F, above 70 F, and above 75 F.*

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A study of these figures will show two things. First, that the variation in energy consumption follows closely the variation in *degree-days above* 65 *F*. The agreement for 1934 is poor, but that for 1935 and 1936 is excellent. Possibly this may be caused by the fact that relatively few customers were operating under this rate in 1934, and consequently the figures for this year cannot be considered conclusive.

In the second place, these figures indicate quite clearly that *degree-days above* 65 *F* give much closer agreement than do degree-days calculated from other base temperatures. As an additional check, similar curves were drawn to a large scale and the degree-days for intermediate temperature bases were included. These additional curves verify the use of 65 F as a base.

No Traceable Relationship with Relative Humidities

The important factor of relative humidity has been neglected entirely in the above investigations. Before proceeding with the use of 65 F as a base, it was considered necessary to investigate the effect of relative humidity. Table 2 has been compiled from U. S. Weather Bureau records and shows the number of days each year during which the outside wet-bulb temperature equalled or exceeded various values.

TABLE 2—DAYS WET-BULB TEMPERATURES WERE EQUALLED OR EXCEEDED MAY 1 TO OCTOBER 15 INCLUSIVE, KANSAS CITY, MO. (HIGHEST OF 7 A.M., NOON, AND 7 P.M. READINGS)

WET-BULB TEMP.	NO. DAYS EQU. OR EXC. WET-BULB TEMP.		
	1934	1935	1936
80	.....	3	.....
79	.....	6	.....
78	.....	12	2
77	8	20	2
76	14	32	10
75	21	39	24
74	32	46	43
73	44	51	54
72	53	59	67
71	70	66	72
70	78	75	78
69	85	82	86
68	91	85	92
67	96	90	102
66	99	97	108
65	109	104	116
64	112	106	119
63	120	108	120
62	127	112	124
61	134	114	132
60	137	119	139

Table 3—Degree-Days Above 65 F, Kansas City, Mo.

Year	May No. Days	June No. Days	July No. Days	August No. Days	September No. Days	October No. Days	Annual Total
1890	75	363	481	268	81	21	1289
1891	336	220	230	276	215	40	1017
1892	28	306	339	359	172	52	1256
1893	48	245	393	264	254	59	1263
1894	99	300	363	410	173	52	1397
1895	131	242	290	322	308	0	1293
1896	169	241	385	390	100	12	1297
1897	79	322	478	309	368	103	1659
1898	102	331	379	368	240	43	1463
1899	124	282	346	446	174	113	1485
1900	122	270	378	470	231	81	1552
1901	66	418	683	440	190	46	1843
1902	174	182	390	349	47	32	1174
1903	84	155	424	309	130	25	1127
1904	52	188	306	285	202	39	1072
1905	60	311	290	411	184	42	1298
1906	147	254	327	393	239	13	1373
1907	58	219	419	367	157	15	1235
1908	140	205	351	332	261	34	1323
1909	52	252	397	481	160	68	1410
1910	30	240	429	323	158	76	1256
1911	224	469	443	404	268	24	1832
1912	179	187	461	380	204	33	1444
1913	120	349	512	633	205	58	1877
1914	138	406	488	410	186	38	1666
1915	82	163	299	183	190	29	946
1916	123	181	569	437	157	46	1513
1917	40	250	462	257	154	20	1183
1918	173	451	424	577	70	58	1753
1919	56	329	524	358	234	53	1554
1920	74	255	368	268	203	71	1239
1921	187	363	498	372	260	29	1709
1922	87	377	373	422	240	68	1567
1923	63	274	445	389	154	5	1330
1924	29	227	302	412	74	76	1120
1925	93	347	423	409	324	21	1617
1926	163	231	431	435	204	19	1483
1927	83	191	372	207	278	71	1202
1928	132	132	418	384	123	91	1280
1929	72	246	421	412	146	20	1317
1930	81	256	550	474	232	65	1658
1931	87	438	519	344	385	85	1858
1932	126	347	507	422	146	25	1573
1933	124	507	491	344	336	19	1821
1934	221	545	721	568	122	62	2239
1935	36	220	675	483	199	31	1644
1936	203	406	706	680	304	35	2334
Totals	4872	13,693	20,480	18,236	9442	2118	68,841
Avg...	104	291	436	388	201	45	1465

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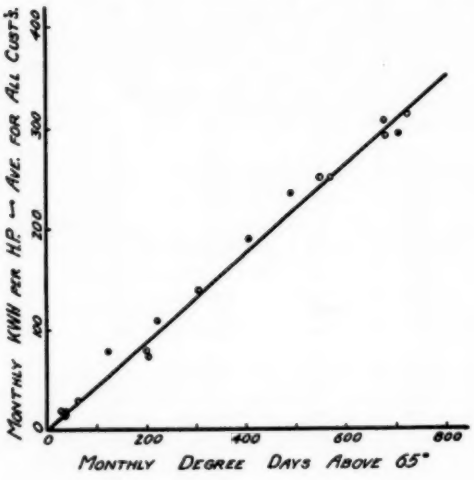

FIG. 10. ENERGY CONSUMPTION VS DEGREE-DAYS

A comparison of Table 2 with Table 1 indicates the absence of any direct traceable relationship. For example in 1934 there were 21 days with 75 F wet-bulb temperatures or higher. In 1935 there were 39, and in 1936, 24 such days. When a wet-bulb temperature of 67 F is considered, the figures are 96 in 1934, 90 in 1935, and 102 in 1936. The energy consumption for commercial customers was 1127 kwhr per horsepower for 1934, 814 in 1935, and 1109 in 1936. These figures indicate the absence of any well-defined relationship between relative humidity and energy consumption.

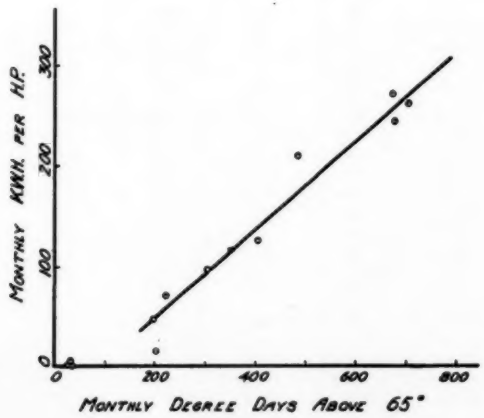

FIG. 11a. ENERGY CONSUMPTION VS DEGREE-DAYS ABOVE 65 F—RESIDENCES

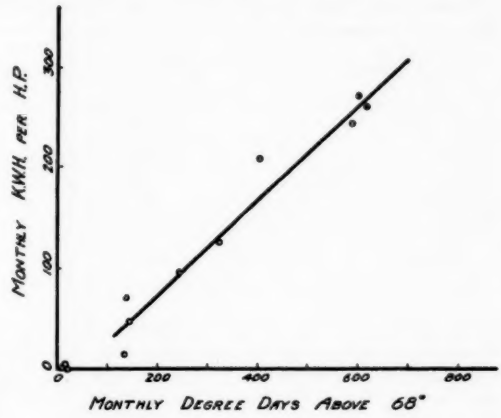

Fig. 11b. Energy Consumption vs Degree-Days Above 68 F—Residences

Degree-Days vs Energy Consumption

Table 3 has been compiled from the records of the U. S. Weather Bureau, and indicates that, for Kansas City, the normal number of *degree-days above* 65 *F* is 1465. It will be noticed that there has been a variation from 946 in 1915 to 2334 in 1936. Incidentally, the average for the past 10 years is 1632, or 11.4 per cent above the 47 year average.

The relation between average energy consumption for all customers and *degree-days above* 65 *F* is given in Fig. 10. From this figure it will be noticed that there is a surprisingly good relation between these two figures where a

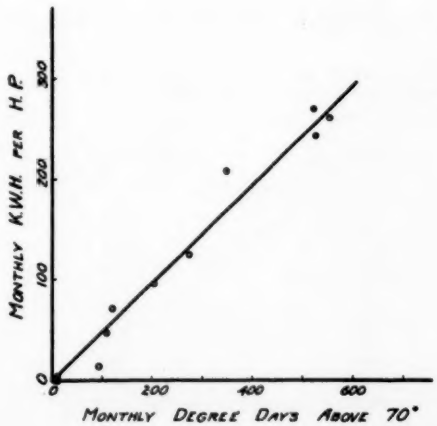

Fig. 11c. Energy Consumption vs Degree-Days Above 70 F—Residences

TABLE 4—ENERGY CONSUMPTION—AIR CONDITIONING, KANSAS CITY, MO

MONTH	NORMAL DEG-DAYS ABOVE 65 F	NORMAL KWHR PER HP	NORMAL PER CENT OF ANNUAL TOTAL
May	104	51	7.4
June	291	137	19.9
July	436	201	29.2
August	388	180	26.1
September	201	97	14.1
October	45	23	3.3
Totals	1465	689	100.0

large group of customers is considered. In order to determine the effect of taking a limited group of customers, Fig. 11a was drawn, using the average consumption of residential customers. In this case the agreement is not as satisfactory, for a line drawn through the points indicates zero consumption for a month with 100 *degree-days above* 65 *F*. Figs 11b and 11c show the same data for *degree-days above* 68 *F*, and *degree-days above* 70 *F*. From these figures it appears that *degree-days above* 70 *F* give better results when residential customers are only considered. For two reasons, however, this cannot be considered as conclusive. In the first place, the number of residences operating more than a year is rather limited. In the second place, those customers who can afford cooling for their homes are in the habit of taking extended summer vacations, during which no cooling is used. This was done in enough cases to throw doubt upon any final conclusions drawn at this time.

Fig. 12 shows that, where commercial customers only are considered, values for *degree-days above* 65 *F* can be used with assurance. This is borne out by Figs. 13 and 14, which show the same data for restaurants and for offices and banks.

From Fig. 10 and Table 3, it is possible to determine the normal energy con-

TABLE 5—ENERGY CONSUMPTION VS HOURS OPEN PER MONTH BASED ON *Normal Year*, KANSAS CITY, MO.[a]

HOURS OPEN PER MONTH	KWHR PER HP PER YEAR	HOURS ACTUAL OPERATION PER YEAR (1 HP = 0.9 KW)
720	1200	1332
660	1120	1120
600	1050	1165
540	975	1083
480	895	994
420	820	911
360	745	828
300	670	744
240	590	656
200	540	600

[a] Compiled by the Kansas City Power & Light Co.

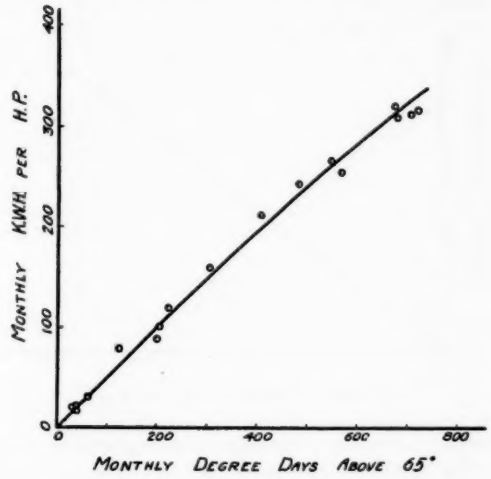

Fig. 12. Energy Consumption vs Degree-Days Above 65 F—Commercial Customers

sumption to be expected each month from a sizable group of air conditioning customers. This information is shown in Table 4, which indicates that a total of 689 kwhr per horsepower can be expected. The distribution of this consumption among the summer months is shown by the last column of Table 4.

Effect of Hours of Operation

The information given in Table 4 is of value to a utility estimating the revenue to be expected from a large and representative group of air condi-

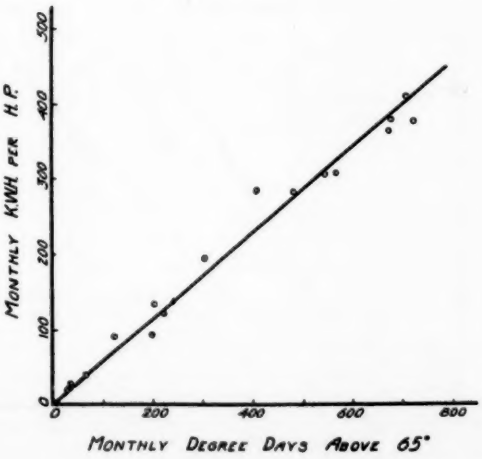

Fig. 13. Energy Consumption vs Degree-Days Above 65 F—Restaurants

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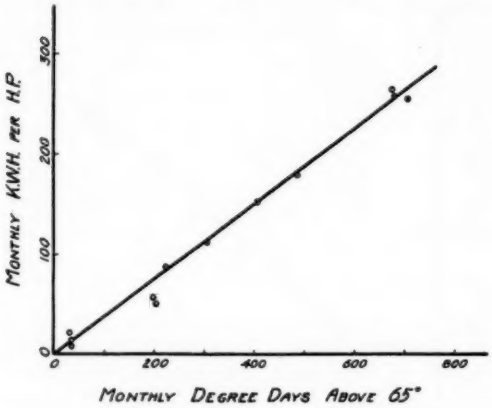

FIG. 14. ENERGY CONSUMPTION VS DEGREE-DAYS ABOVE 65 F—OFFICES AND BANKS

tioning customers. However, it does not give the answer to an engineer desiring to tell one particular customer what will be his operating costs.

In an effort to arrive at values which take into account the class of operation or the hours of operation, Fig. 15 was drawn. In determining the data for this figure, the average number of kilowatt hours per horsepower consumed during 1936 by those customers operating 720 hours a month (24 hours a day) was calculated and shown as one point. Similar points are shown for other classes of operation. Mention should be made of the fact that there are two customers operating 540 hours per month whose operations deviate widely from that of all others. These two have been ignored in drawing Fig. 15.

Through the points determined in this manner was drawn a line showing for 1936 the relationship between hours of operation and annual energy con-

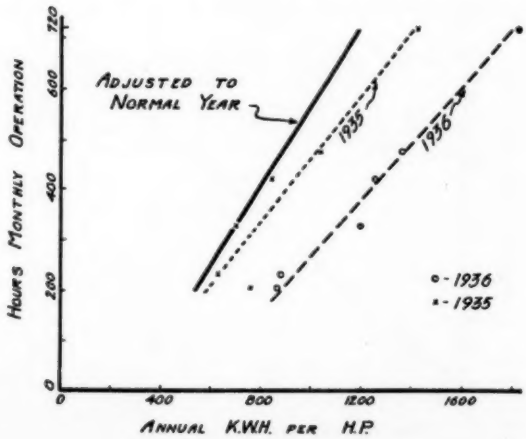

FIG. 15. ENERGY CONSUMPTION VS HOURS OF OPERATION

sumption. A similar line was drawn for 1935. Then these two were adjusted to a normal year in accordance with variations from the normal number of *degree-days above* 65 *F*. The line labeled *Adjusted to Normal Year* is the result.

Table 5 lists in tabular form the points on this line. With this table it is possible to estimate with reasonable accuracy the total annual consumption which any given commercial customer should experience in a normal year.

In using Table 5, it is important to bear in mind the fact that it is based upon the assumption that the cooling plant has been correctly designed. Naturally any deviation from accepted design standards will affect the energy consumption. As an example, in 1935 one restaurant operating 18 hours per day installed a 3-ton compressor, although the calculated heat gain was 7½ tons. From July 4 to August 4, this compressor used 780 kwhr per horsepower—indicating that the motor ran 24 hours per day. This installation was later removed because it was unsatisfactory. As another example, a large downtown restaurant installed 75 hp in 1935. The management of this restaurant insisted on a design that was unnecessarily liberal. During 1935 the energy consumed by this equipment was only 334 kwhr per horsepower—less than half the normal value. This restaurant operates 10 hours per day.

Mention should be made again of the fact that these data cover the operation of equipment which is used only for summer cooling. No attempt was made to include the operation of fans which are used for year-'round air conditioning, although a number of fans used only in the summer months are included. Several of the installations studied have evaporative condensers or cooling towers connected on the same meter as the compressor. Where this was done, it appears that the energy consumption per horsepower for such equipment was substantially the same as that for the balance of the load.

As an example of the use of the information gathered in this investigation, assume that the proprietor of a restaurant operating 300 hours per month desires to know the approximate energy consumption of a 20 hp installation which will be used during summer months only.

From Table 5, it is noted that the total energy consumption should be 20 hp × 670 kwhr or 13,400 kwhr. By referring to Table 4, it is noted that this should be distributed through the year as follows:

May	7.4 per cent × 13,400	990 kwhr
June	19.9 per cent × 13,400	2665 kwhr
July	29.2 per cent × 13,400	3915 kwhr
August	26.1 per cent × 13,400	3500 kwhr
September	14.1 per cent × 13,400	1890 kwhr
October	3.3 per cent × 13,400	440 kwhr
Total		13,400 kwhr

Conclusions

In conclusion it is felt that the investigation described previously indicates the existence of a close relationship between the energy consumption of summer cooling installations and the *degree-days above* 65 *F*. There are some indications that *degree-days above* 70 *F* furnish a better basis for estimates of

energy consumption for residences, but these are not conclusive. It is realized that the information set forth herein applies only to Kansas City, Mo.

The officials of the Kansas City office of the U. S. Weather Bureau have shown a fine cooperative spirit and have indicated their willingness to publish values on *degree-days above* 65 *F* as soon as there is sufficient justification for using this particular temperature. The Washington, D. C., office of the U. S. Weather Bureau has adopted the policy that widespread publication of such values must wait until all investigative studies have been completed and agreement obtained.

Before this relationship can be considered sufficiently established to justify widespread use, additional studies must be undertaken in other parts of the country. In view of the many uses to which this relationship can be applied when once established, it appears that these additional studies should be undertaken without delay. Electric utilities, manufacturers and their distributors, and consulting engineers have all shown an intense interest in the results obtained to date.

In the Kansas City area they are already using them in preparing estimates of operating cost.

DISCUSSION

C. W. WALTON, JR. (WRITTEN): I am pleased to pass along an idea which has been tried at Rockefeller Center and which seems to be giving consistent results. I have not had sufficient time to check through Mr. Marston's paper thoroughly, nor to see how his scheme would work out with our operating data, applied in the manner he suggests.

For the past year tabulated records of performance have been kept, expressed in tons of refrigeration per wet-bulb degree-day. Taking a wet-bulb temperature of 50 F for a base line, it is found that in the case of the large systems, with mixed occupancies, the data obtained are very consistent throughout the range of summer operating conditions. Last season it was necessary for us to work out these wet-bulb averages each day; this year they have been obtainable from the Weather Bureau.

J. DE B. SHEPARD (WRITTEN): Mr. Marston's study is interesting since it represents an attempt to arrive at a general solution of the problem of determining the average expected power consumption of air conditioning plants. However, I feel that before the results can be considered conclusively, a great deal of additional data must be obtained, and great care should be used in applying these general indicated results to specific problems as is now necessary in the use of degree-day figures for heating. It is natural to assume that the power requirements of any air conditioning system will be somewhat affected by varying outside temperatures and humidities. However, in the case of summer air conditioning, the static internal loads represent a very large percentage of the total cooling load and certainly cannot be disregarded. Their effect is probably accounted for to a certain extent by the depression of the base temperature to 65 deg, since experience in this area indicates that outside temperatures must be 65 deg or lower for satisfactory conditions to be maintained in the average commercial premises without the use of refrigeration.

Since dehumidification represents a minor portion of the total cooling load for the average installation, and since there is seldom any attempt made at direct humidity control, it is not surprising that there is no well defined relationship

indicated between absolute humidity and energy consumption for installations in Kansas City. There is probably a fairly well maintained average relationship between dry-bulb and wet-bulb conditions for any locality, which would be taken into consideration in the calculated capacity required for a given job and would not materially affect the relation between hours of use and dry-bulb temperature.

It will be difficult to obtain conclusive information as to the relationship between outside temperature conditions and energy consumption until more information is available relative to costs for complete air conditioned buildings. It is natural to assume that a small air conditioned section of an unconditioned building will be less responsive to changes in outside weather conditions than would the entire structure, since a single section is more or less screened from daily variations by the surrounding structure.

Unfortunately, I have not been able to get information from as many clearly metered cases as has Mr. Marston; however, experience in Baltimore indicates some variation in the power consumption of similar types of enterprises, some of which may be accounted for by the temperament of the persons responsible for the operation. Consumption for an entire season is apparently affected to a greater extent by weather conditions in the late spring or early fall than by variations in the peak months of June, July, August and September. If weather in the late spring is cool, people are not prompted to place their equipment in operation; if the weather in the late spring is warm, equipment is placed in operation and is usually maintained in operation regardless of subsequent cool spells. The reverse is true in the autumn. If the weather early in October is sufficiently cool so that heat is required, many users will have their refrigeration systems pumped back for the season and will not again place the equipment in operation, even though relatively warm weather follows.

These facts are cited merely as an indication that there are many factors now affecting the operating figures being obtained, which are bound to change as people become more accustomed to and more dependent upon comfort cooling. It is interesting to note, however, that, if the human factors affecting the operation of various individual plants are eliminated by averaging the consumption figures for a number of plants, the resulting average consumption falls very close to an estimated consumption figure based upon a study of weather conditions for this territory for a period of 10 years from 1920 to 1930. However, the key average weather figure used in preparing this estimate was the noonday wet-bulb temperature.

I trust that this paper will prompt many others to follow Mr. Marston's footsteps so that additional information relative to this subject may be obtained from other parts of the country.

W. J. KYLE: While Mr. Marston has made it clear that his figures are based on climatic conditions existing at Kansas City, I feel that it is difficult to apply these figures for the country as a whole. However, I am in agreement with the principles outlined. One of the utility companies with whom I am associated has been collecting a large amount of first hand information on the energy required for air conditioning and I feel sure that they would be willing to contribute their information for the benefit of the industry.

A. B. NEWTON:[1] I am wondering if any attempt has been made by the author to break down his data according to the method of operation of systems involved. For instance, I have seen some indication that power requirements for systems which are controlled from both temperature and relative humidity may line up very well with wet-bulb degree-days, while those systems which are controlled from temperature only line up better with dry-bulb degree-days.

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[1] Engrg. Dept., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.

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E. A. WAGNER: It would seem desirable in the study of power used in an air conditioning system to measure and record the quantity and temperature of the water used in this connection.

The water rate would be a big factor in the amount of power used and, if this is not recorded, the conclusions drawn from the results could be very much in error. Both temperature and quantity of water should be tabulated.

A. D. MARSTON: Mr. Shepard has brought out a very excellent point in his discussion of the fact that the static internal loads represent a very large percentage of the total cooling load. This applies primarily to commercial installations. As a matter of fact the increase in tendency toward the use of high intensity lighting is making it necessary to operate commercial air conditioning plants on days when the outside temperature is rather low. There is one extreme example of this in Kansas City, an installation where the very high lighting intensity makes necessary the operation of a cooling plant even in freezing weather.

It seems to me that any study of the power consumption of an air conditioning system must take into account the characteristic difference between residential and commercial installations. In the studies outlined in this paper it was impossible to pursue this point to completion because of the paucity of the records on residential installations.

With reference to Mr. Walton's discussion of the use of a wet-bulb degree-day it does seem that this point is worthy of further study, particularly in the case of large installations where there is both temperature and humidity control.

Mr. Newton asks whether any attempt has been made to break down these data according to the method of operation of the systems involved. The answer is *No* because this information was not specifically compiled. From observation it may be stated that a very large majority of the installations are controlled from temperature only.

Mr. Wagner's point about the quantity and temperature of water used is worthy of consideration in a detailed study. However, it would seem that where a large number of commercial installations is considered this point would average out.

In conclusion it seems to me Mr. Kyle's emphasis on the point that these figures are based only on conditions existing at Kansas City is well taken. It is my hope that this study may stimulate further studies in other parts of the country. It is hoped also that we in Kansas City may be able later to continue these studies to include future data. At the present time we have 196 customers with a connected load of approximately 4000 hp served on this special air conditioning rate. Certainly the data available from such a large group of customers should help in any widespread study of the power consumption by air conditioning systems.

No. 1062

STUDY OF SUMMER COOLING IN THE RESEARCH RESIDENCE USING WATER AT TEMPERATURES OF 52 F AND 46 F

By A. P. KRATZ,* S. KONZO,** AND E. L. BRODERICK † (*MEMBERS*), URBANA, ILL.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *National Warm Air Heating and Air Conditioning Association* and the University of Illinois

THE investigation in summer cooling in the Research Residence during the summer of 1935 [1] made use of water at a temperature of from 58 F to 60 F. The investigation for the summer of 1936 was undertaken to determine to what extent water at temperatures of approximately 52 F and 46 F could be used to produce satisfactory cooling and dehumidification in the Research Residence when supplemented by the circulation of outdoor air through the second story at night and when approximately one air change per hour of outdoor air was used for the purpose of ventilation during the day.

DESCRIPTION OF THE RESEARCH RESIDENCE AND COOLING EQUIPMENT

The Research Residence together with the forced-air heating system has been described in a previous paper.[2] For the purpose of this investigation the Residence was equipped with awnings at all east, south, and west windows, and the sun-room was isolated from the rest of the house by closing the doors leading into the dining room. The entire third story was regarded as an attic and during the daytime was isolated from the rest of the house by closing the door at the head of the stairs. The attic windows, however, were opened to provide ventilation in the attic both day and night. The total space cooled amounted to 14,170 cu ft. Unless otherwise specified, the state of the Residence was the same for the work done during the two summers.

The arrangement [3] of the forced-air duct system and fan was also the same

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† Research Assistant, Engineering Experiment Station, University of Illinois.

[1] A.S.H.V.E. Research Paper, Study of Summer Cooling in the Research Residence Using Water from the City Water Mains, A. P. Kratz, M. K. Fahnestock, S. Konzo, and E. L. Broderick. A.S.H.V.E. JOURNAL SECTION, *Heating, Piping, and Air Conditioning*, May, 1936, pp. 278-288.

[2] A.S.H.V.E. Research Paper, Study of Summer Cooling in the Research Residence at the University of Illinois, A. P. Kratz and S. Konzo, A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 95.

[3] Loc. cit. Note 1.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1937, by S. Konzo.

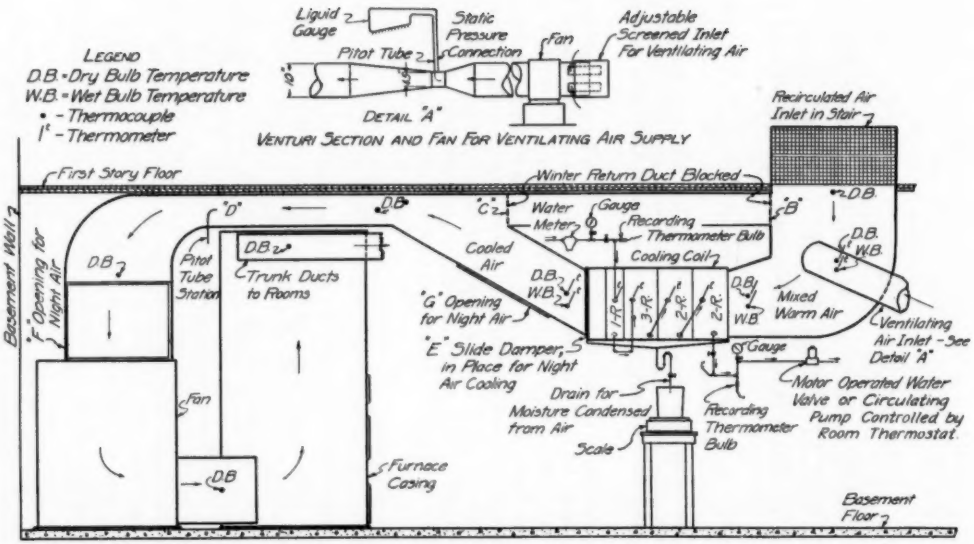

FIG. 1. DIAGRAM OF COOLING PLANT WITH WATER COOLING COIL

for the two summers, and in each case, all return ducts with the exception of the central one containing the cooling coil were blocked. The delivery ducts to the sun-room and third story were blocked, and the dampers in the ducts to the first and second stories were adjusted to maintain the proper balance between the amounts of air required for cooling on these two stories.

The arrangement of the cooling plant is shown in Fig. 1. The cooling coil consisted of 8 rows of finned tubes and one row of tubes without fins. Each row consisted of 12 tubes connected into headers or return bends at the ends of the tubes. These headers were connected in such a way that the water flowed across the duct through one row of 12 tubes and returned through the next row, thus alternating through each succeeding row of tubes. The water connections were made so that the flow of water was counter to the flow of the air, thus providing that the coldest air came in contact with the tubes containing the coldest water. The coil was arranged so that the air flowed horizontally. The water passed through a calibrated water meter before entering the coil. The tubes were ⅝ in. outside diameter and had 8 fins per inch. The overall diameter including the fins was 1⅞₁₆ in. As arranged in the duct, the gross face section of the coil was 17⅝₁₆ in. high and 35½ in. wide, giving a gross face area of 4.33 sq ft. The net face area was 3.56 sq ft. The net free area was 1.76 sq ft. The total area of cooling surface, including the row without fins was 429 sq ft, and, based on a volume of air circulated of 1231 cfm, the gross face velocity was 284, the net face velocity was 346, and the velocity through the net free area was 700 fpm.

For the cooling season of 1935, during which the tests were conducted with water supplied to the coil at a temperature of 58 F, the selection of the cooling coil was based on the following considerations: In selecting the coil required to absorb the maximum cooling load on the Research Residence, it was found that there was considerable latitude in the choice, provided that no restrictions were placed on the amount of cooling water, or the amount of coil surface used. The cooling load was estimated as 20,000 Btu per hour sensible heat and 7,000 Btu per hour latent heat. With 80 F maintained in the rooms, the restriction of 70 F for the temperature of the air leaving the registers, together with the known rise of 4 F in the duct system, determined that the temperature drop of the air passing through the coil should be 15 F; or from 81 F to 66 F. This drop in temperature, together with the sensible heat load, limited the air requirement to 1300 cfm. A net face velocity of 365 fpm permitted the use of an air duct of reasonable dimensions, and thus fixed the face area of the coil. The freedom of choice then proved to be in the amount of coil surface, or the number of rows of coil to be used.

The data furnished by the manufacturer of the coil indicated that the required temperature drop in the air could be obtained either by the use of a large amount of cooling water with a small amount of coil surface, or by a small amount of water used in connection with a large amount of coil surface. Furthermore, the resistance to the flow of air increased rapidly as the amount of surface, or number of rows of coil, increased. In addition to the possible limit imposed on the air delivery by the higher resistance, the use of a large amount of surface has the further disadvantage of high first cost. On the other hand, the use of a small amount of coil surface has the disadvantage of high operating cost due to the amount of water used. Under these conditions a

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compromise is necessary. In the case of the Research Residence, this was effected by selecting 8 rows of coil for the tests with 58 F water. This determined a water rate such that a decrease of one or two rows would have resulted in an appreciable increase in water consumption, while an increase of

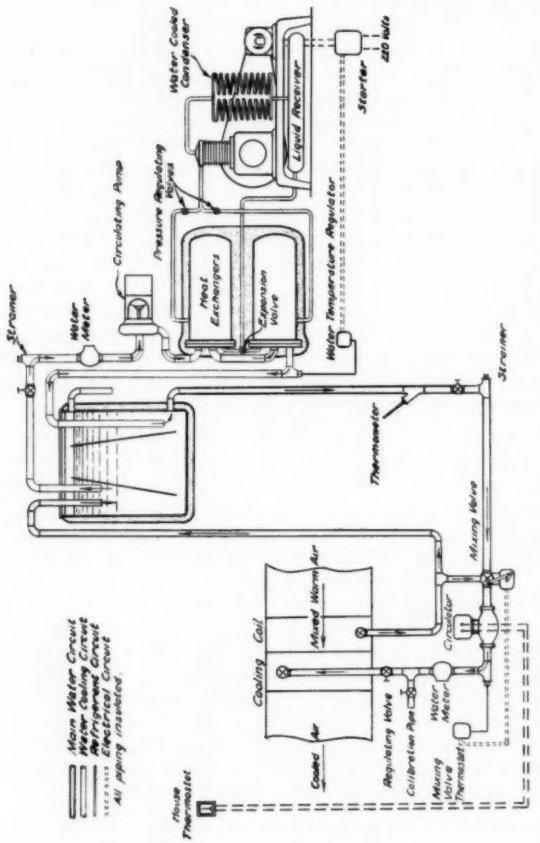

FIG. 2. AUXILIARY EQUIPMENT REQUIRED FOR MAINTAINING DESIRED INLET WATER TEMPERATURE TO COOLING COIL

one or two rows would not have resulted in an appreciable decrease in water consumption.

The resistance offered by the coil itself to the flow of water was only 0.5 lb per square inch. The drop in pressure from the city main to the entrance of the coil, however, was approximately 25 lb per square inch through the ¾ in. line from the main to the coil. This loss in pressure may in some cases be the limiting factor in determining the amount of water available for cooling

purposes. The estimated resistance to flow of air of the 8-row coil was 0.29 in. of water, and this was regarded as the upper limit for the system under consideration.

For the cooling season of 1936, during which the tests were conducted with water supplied to the coil at temperatures of approximately 52 F and 46 F, a similar compromise was effected. The use of the 8 rows of cooling coil previously employed would make possible the use of a small quantity of water while the use of a lesser number of rows of coil would require a large quantity of water. For the latter case the initial cost of the equipment would be lower than for the former case, but the operating cost would be higher as a result of the less economical use of the water. Since a small reduction in the initial cost of the cooling equipment would be offset over a period of years by the reduced annual operating cost, it was considered advisable to retain the same number of coil sections used in the tests with 58 F water and to make suitable

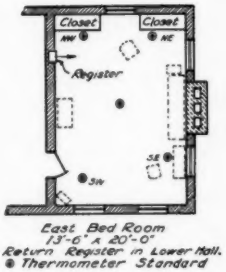

FIG. 3. LOCATION OF REGISTER WITH RESPECT TO EXPOSED WALLS

reductions in the quantity of water used. Hence, the tests for the season of 1936 were made with rates of water flow of approximately 260 gal per hour and 165 gal per hour with water temperatures of 52 F and 46 F, respectively, as compared with the 360 gal per hour used in connection with the 58 F water.

The operation of the cooling plant was controlled by means of a thermostat located at the 60-in. level in the hall on the second story. As shown in Fig. 2 this thermostat was used in connection with a motor-operated, water circulator which was placed on the inlet side of the cooling coil and which was used to start and stop the flow of water through the coil. The temperature of the water entering the coil was maintained at a constant value by means of the three-way mixing valve located on the inlet side of the circulating pump. This valve served to proportion and mix the warm water received from the outlet of the cooling coil and the cooled water received from the heat exchangers so that the temperature of the water entering the cooling coil was maintained at a predetermined value of either 52 F or 46 F. The operation of the three-way valve was automatic, and was controlled by means of a regulating thermostat placed in the water circuit on the outlet side of the water circulator, as indicated in Fig. 2.

For the tests conducted in 1936, since water from the city water mains was not available at temperatures of less than 58 F, it was necessary to provide

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a reservoir of water that was cooled to a predetermined temperature slightly lower than the temperature of water entering the cooling coil. The auxiliary equipment which was required to maintain the proper temperature in the reservoir is shown in the right hand portion of Fig. 2. It may be observed that the warm water which left the cooling coil was passed through heat exchangers by means of a circulating pump, was cooled, and was then returned to the reservoir from which the cold water supply to the cooling coil was obtained. The heat exchangers consisted of two shell-and-tube type condensing units in which the warm water passed through the tubes and the refrigerant was circulated over the outside of the tubes. The auxiliary equipment provided a very satisfactory means of obtaining the desired quantity and temperature of water for the cooling coil and was an excellent substitute for a natural cold water supply.

In order to provide for both heating in the winter and cooling in the summer the cooling coil was installed in a by-pass in the central return duct as shown in Fig. 1. For the summer work, the duct was blocked with tightly fitting dampers at *B* and *C*, and all of the air delivered by the fan in the forced-air heating system passed through the cooling coil when the air in the house was being recirculated. For the purpose of providing outdoor air for cooling during the night, a slide damper, *E*, was placed in the by-pass on the outlet side of the coil, and a door, *G*, was placed in the recirculating duct on the downstream side of the slide damper. When outdoor air was required, the basement door and the door, *G*, in the recirculating duct were opened, and the slide damper, *E*, was closed. The fan in the forced-air system delivered approximately 1231 cfm of air when recirculating the air in the house and 2220 cfm when using outdoor air at night. These air volumes were based on an air density of 0.0749-lb per cubic foot. Outdoor air, for the purpose of ventilation during the day, was provided by means of the duct shown as Detail *A* in Fig. 1.

The study of baseboard and sidewall registers was made in the east bedroom of the Research Residence. This room was 13½ ft by 20 ft, with an 8 ft-4 in. ceiling, and had three exposed walls; north, south, and east. It had one door, which remained open during all of the tests.

The location and arrangement of the warm-air stack and registers are shown in Figs. 3 and 11. The sidewall register was the same size as the baseboard register and was placed at the head of the extension of the stack for the baseboard register in order to facilitate changes from one position to the other. The same register face was used in both locations, and consisted of horizontal louvers, 1 in. in width, which could be set at any angle from 0 deg to 90 deg with the horizontal. In the latter case the louvers were completely closed. When the sidewall register was in use, the baseboard outlet was sealed with a flat plate placed with the inner face flush with the wall of the stack. Similarly, when the baseboard register was in use, the stack was sealed just above the baseboard outlet. Hence, in either case, the stack was continuous and without offsets or projections.

METHOD OF CONDUCTING TESTS

Continuous records were made during the summer of air temperatures in the various parts of the house and in the different sections of the duct system.

Other incidental temperatures and relative humidities were also read at regular intervals. A complete description of the measurements which were made has been given in the previous report.[4]

During the periods of operation, observations were made of the weight of water circulated through the cooling coil, the temperature of the water entering and leaving the coil, the weight of water condensed from the air, and the electrical input to the fan motor. The weight of water circulated was obtained by means of a calibrated water meter and the air quantities were obtained from traverses made with Pitot tubes at section *D* in the central return duct and at the venturi section in the ventilating air duct as shown in Fig. 1. Incidental data from the operation of the heat exchangers were also obtained.

For all of the tests the Residence was operated on the following schedule:

The second story windows and the attic and basement doors were closed at 7 a.m. and the dampers were changed so that the fan, which had been delivering outdoor air through the system started delivering recirculated air and the outdoor air admitted for ventilation. The former was equivalent to 4.2 recirculations of the air in the house and the latter was equivalent to one air change per hour, making a total of 5.2 air changes per hour delivered by the fan. The fan was allowed to run continuously during the day. When the temperature of the indoor air on the second story rose to 81 F the motor driven water circulator, actuated by the thermostat, admitted water to the cooling coil. The cooling plant was allowed to operate with thermostatic *on* and *off* control maintaining 81 F on the second story until the effective temperature outdoors became equal to the effective temperature on the second story indoors. The water circulator was then stopped; eleven of the windows on the second story and the attic door were opened; and the dampers were set and the basement door opened, so that the fan delivered outdoor air through the duct system, the fan continuing to run until 7 a.m. The fan delivery was 2220 cfm or 9.4 air changes per hour.

In the first series of tests, which has been designated as Series 1-36, the temperature of the water which was supplied to the cooling coil was maintained at approximately 52 F and the quantity of water circulated through the coil was regulated so that the flow was approximately 260 gal per hour. In the second series of tests, designated as Series 2-36, the water temperature was maintained at approximately 46 F and the quantity of the water flowing was adjusted to a value of approximately 165 gal per hour.

During all tests made in the study of baseboard and sidewall registers, the cooling plant was operated normally to maintain approximately 80 F at the 5 ft level in the rooms of the Residence. Observations were then made of the velocity and temperature of the air leaving the register, and of the air temperature on the central axis of the room 4 in. above the floor, 5 ft above the floor, and 4 in. below the ceiling. In addition, for some of the tests the air temperatures at two levels, 4 in. above the floor and 5 ft above the floor, were observed in all the four corners of the room, as shown in Fig. 3. For each register arrangement a large number of observations was made to include a wide range of outdoor temperatures. The temperatures at the different levels were plotted against the difference between the outdoor temperature and the temperature at the 5 ft level in the room.

[4] Loc. cit. Note 1.

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TABLE 1—TYPICAL OPERATING DATA AND RESULTS OBTAINED BY COOLING WITH WATER AT 2:00 P.M. ON JULY 7 AND JULY 13, 1936

No.			
1	Date	7–7–36	7–13–36
2	Test No.	12	18
3	Series	1–36	2–36
4	Outdoor Air		
	Dry Bulb, F	103.8	103.8
	Wet Bulb, F	75.8	76.8
	Dew Point, F	63.8	65.7
	Relative Humidity, per cent	27.8	29.5
	Specific Humidity, grains per pound dry air	88.4	96.2
5	Indoor Air, average br. level temp.		
	1st Story, F	79.4	79.5
	2nd Story, F	81.6	80.8
6	Indoor Air, house average at br. level		
	Dry Bulb, F	80.3	80.1
	Wet Bulb, F	67.6	67.4
	Dew Point, F	61.0	61.0
	Relative Humidity, per cent	51.8	52.0
	Specific Humidity, grains per pound dry air	80.8	80.5
7	Indoor Air, entering return grille		
	Dry Bulb, F	78.9	78.8
	Wet Bulb, F	67.3	66.8
	Dew Point, F	61.5	60.5
	Relative Humidity, per cent	55.0	53.3
	Specific Humidity, grains per pound dry air	82.0	78.5
8	Ventilating Air		
	Dry Bulb, F	97.8	99.0
	Wet Bulb, F	72.8	71.4
	Dew Point, F	61.0	57.6
	Relative Humidity, per cent	30.0	25.6
	Specific Humidity, grains per pound dry air	80.7	71.0
9	Mixed Air, entering cooling coil		
	Dry Bulb, F	81.0	81.4
	Wet Bulb, F	67.1	66.6
	Dew Point, F	59.8	58.7
	Relative Humidity, per cent	47.7	45.2
	Specific Humidity, grains per pound dry air	77.0	73.8
10	Mixed Air, leaving cooling coil		
	Dry Bulb, F	64.0	63.9
	Wet Bulb, F	60.8	60.0
	Dew Point, F	58.9	57.7
	Relative Humidity, per cent	83.3	79.9
	Specific Humidity, grains per pound dry air	74.1	71.0
11	Air temp. drop through cooling coil, F	17.0	17.5
12	Temp. cooled air leaving registers, average, F	67.7	67.7
13	Air temp. rise in ducts and casing, F	3.7	3.8
14	Basement air temp. at br. level, F	74.0	74.2
15	Quantity of air through wet coil		
	Cfm	1231	1231
	Pounds	5540	5540
	Density of air, pounds per cubic foot	0.075	0.075
16	No. of house air recirculations per hour	5.2	5.2
17	Ventilating Air		
	Cfm	226	226
	Pounds	1016	1016
	Density of air, pounds per cubic foot	0.075	0.075

TABLE 1 (*Continued*)—TYPICAL OPERATING DATA AND RESULTS OBTAINED BY COOLING WITH WATER AT 2:00 P.M. ON JULY 7 AND JULY 13, 1936

No.			
18	Cooling Coil		
	Gross Face Area, sq ft	4.33	4.33
	Net Free Area, sq ft	1.76	1.76
	Surface of Coil, sq ft	429	429
19	Air Velocity through Coil		
	Gross Face, fpm	284	284
	Net Face, fpm	346	346
	Free Area, fpm	700	700
20	Moisture condensed from air, pounds per hour	2.26	2.24
21	Heat given up by air; total Btu per hour	24,970	25,630
	Heat due to moisture in air		
	Btu per hour	2370	2350
	Per cent of total heat	9.5	9.2
	Sensible heat		
	Btu, per hour	22,600	23,280
	Per cent of total heat	90.5	90.8
22	Water temp. in cooling coil		
	Inlet, F	51.9	46.9
	Outlet, F	63.4	65.4
23	Rise in water temp. through coil		
	First row, F	1.1	2.6
	First four rows, F	5.3	9.7
	First six rows, F	8.2	13.8
	Total coil, eight rows, F	11.5	18.5
24	Water pressure at coil, pounds per sq in.	4.5	4.5
25	Quantity of cooling water		
	Pounds per hour	2100	1344
	Gallons per hour	252	161
26	Heat absorbed by water, Btu per hour	24,150	24,860
27	Recirculating fan speed, rpm	445	445
28	Fan motor data		
	Horse power	¾	¾
	Power rate, watts	440	440
29	Static pressure loss through coil		
	Coil, wet, in. water	0.28	0.28
	Coil, dry, in. water	0.23	0.23
30	Total resistance of system, in. water	0.51	0.51

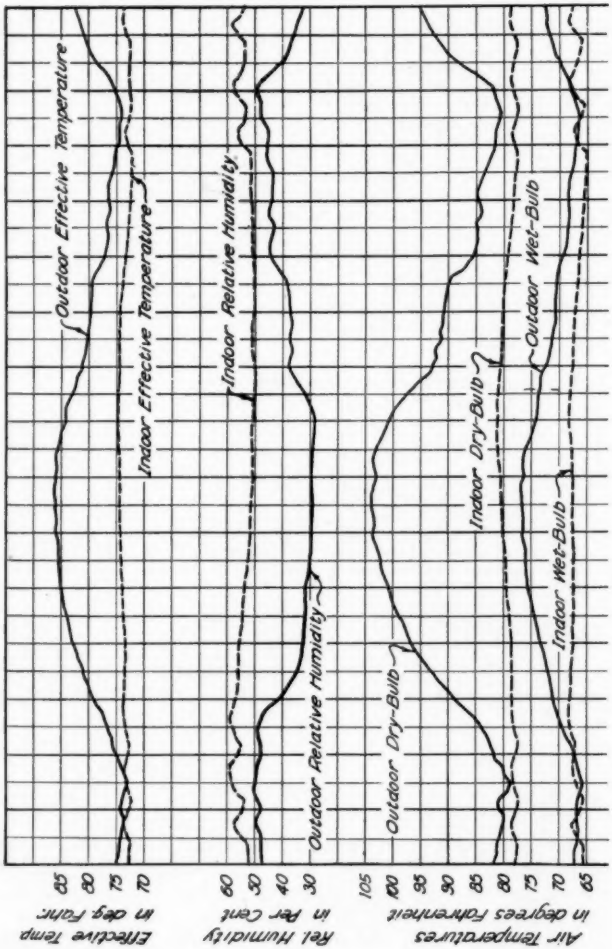
RESULTS OF TESTS

General Conditions

The operating characteristics of the cooling plant and a comparison of the actual and calculated cooling loads for the house can best be illustrated by the results obtained on a typical day. For this purpose the test made on July 7, 1936, was selected as typical of the operation obtained with water at a temperature of 52 F (Series 1-36) and the results are shown in the first column of Table 1 and in Fig. 4. The test made on July 13, 1936, was selected as typical of the operation obtained with water at a temperature of 46 F (Series 2-36) and the results are shown in the second column of Table 1. General results from all the tests are given in Table 2.

On the day which was selected as being typical for the tests made under

conditions of operation for Series 1-36, the outdoor temperature was 103.8 F at 2 p.m. and reached a maximum of 104.0 F at 3 p.m. At this time the house was being operated with a constant dry bulb temperature of approximately 80



F. On the days preceding this test the outdoor air temperature had attained a maximum value of 100 F, 102 F, and 105 F. During the greater portion of the early morning hours of July 7, 1936, the outdoor temperature was greater than 80 F and reached a minimum value of 78.5 at 5 a.m. Since the outdoor

air temperature was much greater than the indoor air temperature on the night previous to the test the windows remained closed and no outdoor air, other than the amount required for ventilation, was introduced into the house during the night. The air conditions at different locations are given in items 4 to 14 of Table 1. The dry-bulb temperature on the second story was 2.2 F higher than that on the first story, indicating a slight unbalance in the cooling on the two stories. It was observed that this difference between the average temperatures of the first and second story rooms was of the order of only 0.5 F for a day in which the maximum outdoor temperature was 95 F, but increased to approximately 2.5 F for a day in which the maximum outdoor temperature was 108 F. This indicated that when the outdoor temperature was excessively high the cooling requirement for the second story became a relatively greater proportion of the total cooling requirement for both stories. The slight unbalance in the cooling load on the two stories during the few extremely hot days was not accompanied by any particular difficulties either in the conditions of operation or in the comfort maintained in the Residence. The temperature of the cooled air leaving the registers was approximately 68 F. The results obtained from tests made with three different arrangements of registers in the east bedroom are presented in a subsequent section of this paper. The rise in temperature of the air passing through the furnace casing and ducts was 3.7 F. The quantity of air circulated was 1265

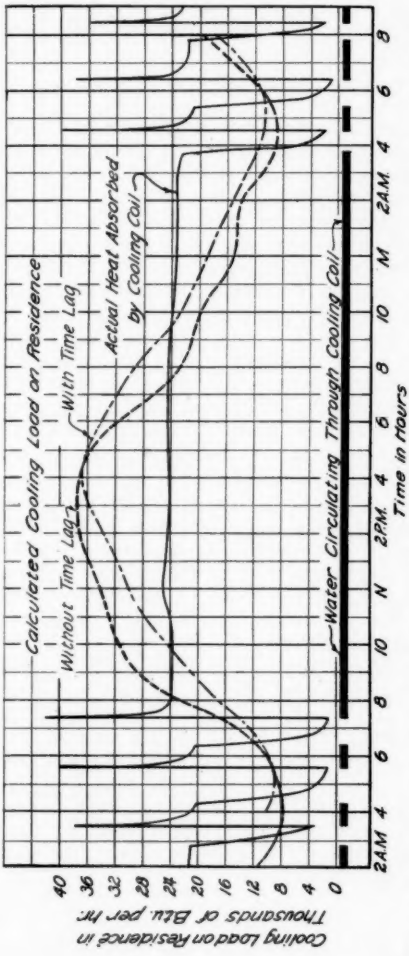


FIG. 4. ACTUAL AND CALCULATED COOLING LOAD ON RESIDENCE AND AIR TEMPERATURES. TEST NO. 12, SERIES 1-36, JULY 7, 1936. 52 F COOLING WATER

cfm when the coils were dry. Increased frictional resistance when condensation was present on the coil surfaces, however, decreased this amount to 1231 cfm. Since part of the coil surfaces were always wet during actual operating periods, all calculations were based on 1231 cfm for the volume of air circulated at a density of 0.0749 lb per cubic foot. The free area velocity of 700 fpm did not prove to be sufficient to carry any condensed moisture away from the surfaces of the cooling coil.

Heat Balance

Since the cooling coil was well lagged with corkboard to prevent heat gain, it was possible to obtain the actual cooling load either from the heat given up by the mixture of air and water vapor or from the heat absorbed by the water passing through the coil. However, the cooling studies made in 1935 had indicated that unless complete temperature traverses were made in the duct on both sides of the coil and proper corrections were applied to the averages of the temperatures obtained from the wet- and dry-bulb thermometers at the two reading stations on each side of the coil, the discrepancy in heat balance obtained by the two methods was very large. By applying proper corrections to the dry-bulb and wet-bulb reading, however, and using the corrected values of the latter to determine the enthalpy of the moist air entering and leaving the coil, it was possible to obtain a heat balance between the air and water within approximately 5 per cent. This method of calculation, however, does not show the distribution between sensible and latent heat loads. This distribution was obtained, as shown in Item 21 Table 1, by using the corrected values for the dry-bulb temperatures, and by multiplying the weighed amount of condensation from the coil by a constant representing the heat given up by the change in moisture content of the air per pound of water vapor condensed. This latter included the latent heat and superheat in the water vapor condensed, and the change in the superheat of the water vapor remaining in the air after passing the cooling coil, and amounted to 1,050 Btu per pound of vapor condensed. The total of the heat given up by the air as computed by this method was within 4 per cent of the calculated heat absorbed by the water. For the purpose of analysis, and in the general results shown in Table 2, the cooling load calculated from the water circulated through the coil, as shown in Item 26, Table 1, was accepted as correct. The latent heat load was calculated from the weighed amount of moisture condensed from the air and the sensible heat load was obtained by difference. As shown in Item 21 in Table 1, the moisture load for the day under consideration was 9.5 per cent of the total load. It may also be observed from column 34 in Table 2 that for the tests made during the season of 1936 the moisture load varied from 0 to 21 per cent of the total load. A more detailed discussion of this ratio of moisture load to the total load is presented in a subsequent section of this paper.

Operating Characteristics

The operating characteristics with 52 F cooling water and with a constant dry-bulb temperature in the rooms of approximately 80 F are shown in Fig. 4, for a day on which the outdoor temperature rose to a maximum of 104 F. The house had remained closed for several days previous to this test and the cooling plant had been operating intermittently to maintain a constant dry-bulb temperature in the rooms both day and night. On this test the plant

operated intermittently from 2:50 a.m. to 7:20 a.m. and continuously from 7:20 a.m. to 3:50 a.m. maintaining the dry-bulb temperature between 78 F and 81.5 F.

At 5:35 a.m. after the water circulation had been stopped about 80 min, the wet-bulb temperature was 67.8 F corresponding to a relative humidity of 59 per cent. When the water circulation was started the relative humidity decreased to 56 per cent, but increased again to 59 per cent almost immediately after the water circulation was stopped. This alternate decrease and increase of relative humidity in the house was characteristic of the intermittent operation of the plant.

It is of interest to note that in the summer of 1935 when the outdoor conditions were such that the air contained more moisture than it did in 1936, which was an excessively dry season, the total variation in the indoor relative humidity during the on- and off-periods of plant operation was comparatively large, amounting to approximately 13 per cent. It is probable that if the 1935 season had been more normal from the standpoint of outdoor humidity, the magnitude of the variation in indoor humidity, resulting from intermittent operation of the plant, would have been much smaller than 13 per cent, but it is hardly probable that it would have been as small as the 3 per cent shown in Fig. 4. It is also possible that some of the difficulties that were experienced in 1935 when the cooling plant was operated intermittently in order to maintain a constant dry-bulb temperature might have been avoided if the outdoor air had contained less moisture. It is apparent that the indoor relative humidity that exists during the off-periods of any intermittently operated cooling unit is largely dependent on the prevailing outdoor humidity conditions. During the season of 1936 because of the very small changes in the relative humidity occurring between the on-periods and the off-periods of the cooling plant, and resulting as a consequence of the excessively dry season, the conditions maintained in the house were entirely satisfactory even for those days on which the plant was operating intermittently over a large portion of the time.

When the plant operated continuously, as from 7:20 a.m. to 3:50 a.m., the relative humidity stabilized at between 50 and 55 per cent, representing an effective temperature of approximately 74 deg. During the period between 4 p.m. and 8 p.m. when the average indoor dry-bulb temperature was approximately 81 F, the relative humidity in the house was approximately 51 per cent and the corresponding effective temperature was 74.5 deg. In spite of the fact that on these very hot days the capacity of the cooling plant was not quite sufficient to prevent the indoor temperatures from exceeding 80 F, particularly on the second story, the effective temperature did not exceed 75 deg at any time. Hence the indoor conditions, both during the periods of intermittent operation and during the periods of continuous operation of the cooling plant, were on the upper borderline for comfort and could be regarded as entirely satisfactory. As indicated from the values listed in column 2 of Table 1, the indoor conditions that were maintained with the water entering the cooling coil at 46.9 F were practically identical with those maintained when the temperature of the entering water was 51.9 F. That is, in both series of tests, the indoor dry-bulb temperature, the indoor relative humidity, and the indoor effective temperature respectively were maintained at approximately the same values, irrespective of the temperature of the water entering the coil.

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TABLE 2—COMPLETE DATA AND RESULTS FOR TESTS

DATE 1936	TEST No.	SERIES No.	WEATHER DATA							INDOOR AIR CONDITIONS			
			Degree-Hours		Sun Hours: Minutes	Outdoor Air				At End of Night Cooling		Average During Period Requiring Cooling with Water	
						Maximum Temp. During Test F	Minimum Temp. Previous to Test F	Average During Period Requiring Cooling with Water					
			Above 85°	Above 90°				Temp. F	Hum. %	Temp. F	Hum. %	Temp. F	Hum. %
1	2	3	4	5	6	7	8	9	10	11	12	13	14
6–17–36	1	1–36	66.5	25.4	11:05	95.0	67.0	88.2	51.7	74.2	52.0	78.8	48.1
6–20–36	2	1–36	47.0	10.2	15:07	94.0	68.0	84.8	49.8	73.7	55.1	79.1	49.9
6–27–36	3	1–36	19.1	0.0	5:10	89.0	69.0	83.8	54.8	76.0	46.9	77.9	53.5
6–28–36	4	1–36	43.8	6.4	13:20	93.0	68.0	88.8	36.8	76.7	62.6	79.1	49.3
6–29–36	5	1–36	137.0	68.9	8:15	101.0	60.0	87.4	44.1	70.8	40.8	78.7	52.4
7– 2–36	7	1–36	28.9	0.0	10:46	92.0	67.0	85.0	30.4	73.8	63.3	79.2	48.5
7– 3–36	8	1–36	34.8	0.3	15:06	91.0	64.0	86.7	37.1	73.7	66.5	79.0	53.0
7– 4–36	9	1–36	125.0	63.3	15:02	100.0	72.0	88.4	41.3	76.9	64.0	79.0	54.5
7– 5–36	10	1–36	142.1	75.5	13:47	102.0	73.0	90.8	40.2	....	....	78.8	55.4
7– 6–36	11	1–36	174.3	103.0	14:53	105.0	79.0	91.6	38.6	....	....	79.5	54.6
7– 7–36	12	1–36	194.5	117.8	14:25	104.0	78.0	93.2	36.7	....	....	79.8	52.6
7– 8–36	13	1a–36	170.2	107.2	14:59	104.0	80.0	97.2	32.8	....	....	80.3	51.2
7– 9–36	14	1a–36	93.9	44.0	11:34	99.0	72.0	87.8	43.1	79.0	61.8	78.9	56.5
7–10–36	15	2–36	163.6	97.3	10:34	104.0	76.0	91.3	39.9	....	....	79.2	53.3
7–11–36	16	2–36	184.3	111.7	11:48	105.0	79.0	92.8	37.4	....	....	79.5	52.9
7–12–36	17	2–36	203.6	125.5	13:52	106.0	79.0	92.0	39.1	....	....	79.7	52.3
7–13–36	18	2–36	180.9	104.2	12:47	105.0	75.0	92.4	38.9	....	....	79.7	54.2
7–14–36	19	2–36	207.6	135.4	14:88	108.0	79.0	93.0	35.6	....	....	79.7	50.5
7–15–36	20	2–36	93.0	57.4	5:05	107.0	77.0	86.6	49.5	....	....	78.8	57.9
7–16–36	21	2–36	83.1	31.4	14:50	98.0	74.0	90.2	40.1	....	....	78.9	56.1

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7–17–36	22	2–36	108.9	52.6	14:48	100.0	68.0	91.6	31.3	75.9	58.4	78.8	50.2
7–18–36	23	2–36	0.9	0.0	6:00	92.0	75.0	83.5	44.1	77.4	76.9	79.1	58.7
7–23–36	24	2–36	98.6	47.5	11:51	97.0	73.0	92.4	36.9	76.8	71.1	79.4	56.8
7–24–36	25	2–36	8.0	0.0	11:18	92.0	67.0	83.3	41.4	75.5	63.7	79.2	56.3
7–25–36	26	2–36	18.1	0.0	9:00	90.0	66.0	84.6	62.2	74.1	76.6	79.3	71.9
7–26–36	27	2–36	98.3	48.0	12:23	99.0	76.0	88.0	47.0	79.0	72.1	79.3	60.6
7–27–36	28	2–36	107.0	60.5	12:10	101.0	77.0	93.6	38.3	....	....	79.7	54.4
7–28–36	29	2–36	33.2	4.7	11:22	93.0	70.0	87.8	43.1	77.1	72.9	79.5	59.3
8– 3–36	30	2–36	12.1	0.0	10:26	87.0	68.0	82.9	48.1	74.8	66.5	79.8	59.5
8– 9–36	31	2–36	50.4	10.5	14:02	93.0	68.0	86.1	46.3	71.5	58.8	79.3	59.8
8–10–36	32	2–36	5.6	0.0	11:41	87.0	67.0	83.9	46.2	75.3	73.0	79.5	64.4
8–13–36	33	2–36	88.3	35.2	13:30	97.0	71.0	88.9	45.1	75.8	69.1	79.1	59.1
8–14–36	34	2–36	18.8	0.6	6:06	90.0	71.0	82.7	50.1	76.9	72.8	79.6	63.9
8–15–36	35	2–36	68.7	23.0	13:51	95.0	77.0	84.3	53.6	....	....	78.9	64.1
8–16–36	36	2–36	98.4	16.2	13:48	95.0	72.0	88.2	49.4	....	....	79.3	63.5
8–17–36	37	2–36	21.6	0.0	13:46	89.0	68.0	84.2	42.2	75.1	66.5	79.5	57.3
8–18–36	38	2–36	130.3	66.5	13:45	101.0	72.0	89.9	41.6	76.2	63.0	79.2	56.9
8–19–36	39	2–36	94.5	36.3	12:42	95.0	80.0	91.5	43.1	....	....	79.1	56.6
8–20–36	40	1–36	53.4	15.4	9:15	95.0	70.0	85.9	45.4	76.8	77.5	79.2	61.0
8–21–36	41	1–36	115.3	54.5	13:07	99.0	79.0	88.4	45.6	....	....	78.9	59.3
8–22–36	42	1–36	126.0	63.4	13:35	99.0	76.0	88.6	44.1	....	....	79.0	58.4
8–23–36	43	1–36	66.5	24.1	8:34	96.0	74.0	88.4	45.6	....	....	78.8	59.8
8–24–36	44	1–36	82.0	34.8	13:20	96.0	71.0	86.2	48.3	77.3	70.6	79.1	61.0
8–25–36	45	1–36	86.2	34.1	12:07	95.0	74.0	90.2	42.5	....	....	79.3	57.6
8–26–36	46	1–36	72.6	32.2	11:54	98.0	66.0	82.7	58.2	75.2	76.0	79.1	67.0
8–27–36	47	1–36	7.2	0.0	7:27	88.0	71.0	84.6	49.4	....	....	79.3	62.6
8–28–36	48	1–36	14.2	0.0	8:25	89.0	69.0	81.9	52.6	75.2	79.3	79.7	67.3
9–11–36	49	1–36	48.4	9.2	11:00	94.0	68.0	82.7	54.7	71.0	78.8	79.2	64.1
9–14–36	50	1–36	37.4	2.2	12:35	91.0	71.0	85.2	50.8	75.9	76.1	79.7	67.6
9–15–36	51	1–36	26.0	0.0	12:32	90.0	70.0	84.1	49.4	76.0	75.0	79.7	66.2

TABLE 2 (*Continued*)—COMPLETE DATA AND RESULTS FOR TESTS

DATE 1936	TEST No.	SERIES No.	TEMPERATURE DIFFERENCE		OPERATION OF THE FAN						OVERALL TEST PERIOD		
					Night Air Cooling Previous to Test		Recirculation Previous to Test		Recirculation During Test Period		Start of Cooling with Water to Start of Night Air Cooling		
			Outdoor Max. to Average Indoor F	Average Outdoor to Average Indoor F	Duration Hours	Power Kwhr	Duration Hours	Power Kwhr	Duration Hours	Power Kwhr	Start	End	Duration Hours
1	2	3	15	16	17	18	19	20	21	22	23	24	25
6–17–36	1	1–36	16.2	9.4	8.50	4.4	8.50	3.7	6.25	2.7	3:30p	9:45p	6.25
6–20–36	2	1–36	14.9	5.7	9.33	4.7	8.25	3.6	4.25	2.0	3:15p	7:30p	4.25
6–27–36	3	1–36	11.1	5.9	10.67	5.3	9.48	4.1	6.85	3.2	4:30p	11:30p	6.85
6–28–36	4	1–36	13.9	9.7	7.50	3.7	5.05	2.2	8.00	3.6	12:03p	8:03p	8.00
6–29–36	5	1–36	22.3	8.7	10.95	5.5	9.23	4.1	16.77	7.5	4:14p	9:00a	16.77
7– 2–36	7	1–36	12.8	5.8	12.67	6.5	11.08	4.8	1.92	0.7	6:05p	8:00p	1.92
7– 3–36	8	1–36	12.0	7.7	11.00	5.5	9.03	4.0	4.47	2.0	4:02p	8:30p	4.47
7– 4–36	9	1–36	21.0	9.4	10.50	5.2	4.25	1.8	21.75	9.6	11:15a	9:00a	21.75
7– 5–36	10	1–36	23.2	12.0	0.00	0.0	0.00	0.0	23.90	10.4	9:00a	9:00a	24.00
7– 6–36	11	1–36	25.5	12.1	0.00	0.0	0.00	0.0	24.00	10.7	9:00a	9:00a	24.00
7– 7–36	12	1–36	24.2	13.4	0.00	0.0	0.00	0.0	24.00	10.4	9:00a	9:00a	24.00
7– 8–36	13	1a–36	23.7	16.9	0.00	0.0	0.00	0.0	13.00	5.9	9:00a	10:00p	13.00
7– 9–36	14	1a–36	20.1	8.9	9.00	4.5	1.00	0.4	25.00	11.0	8:00a	9:00a	25.00
7–10–36	15	2–36	24.8	12.1	0.00	0.0	0.00	0.0	24.00	10.7	9:00a	9:00a	24.00
7–11–36	16	2–36	25.5	13.3	0.00	0.0	0.00	0.0	24.00	10.5	9:00a	9:00a	24.00
7–12–36	17	2–36	26.3	12.3	0.00	0.0	0.00	0.0	24.00	10.7	9:00a	9:00a	24.00
7–13–36	18	2–36	25.3	12.7	0.00	0.0	0.00	0.0	24.00	10.4	9:00a	9:00a	24.00
7–14–36	19	2–36	28.3	13.3	0.00	0.0	0.00	0.0	24.00	10.6	9:00a	9:00a	24.00
7–15–36	20	2–36	28.2	7.8	0.00	0.0	0.00	0.0	23.97	10.3	9:00a	9:00a	24.00
7–16–36	21	2–36	19.1	11.3	0.00	0.0	0.00	0.0	14.00	6.1	9:00a	11:00p	14.00

7–17–36	22	2–36	21.2	12.8	8.00	4.0	5.67	2.6	11.33	5.0	12:40p	12:00Mid	11.33
7–18–36	23	2–36	12.9	4.4	7.00	3.2	8.32	3.6	4.68	2.1	3:19p	8:00p	4.68
7–23–36	24	2–36	17.6	13.0	9.90	4.9	4.68	2.0	11.57	5.1	11:41a	11:15p	11.57
7–24–36	25	2–36	12.8	4.1	7.75	3.8	7.47	3.2	5.03	2.1	2:28p	7:30p	5.03
7–25–36	26	2–36	10.7	5.3	11.50	5.7	9.98	4.3	6.35	2.8	4:59p	11:20p	6.35
7–26–36	27	2–36	19.7	8.7	7.67	3.8	2.18	1.0	23.82	10.2	9:11a	9:00a	23.82
7–27–36	28	2–36	21.3	13.0	0.00	0.0	0.00	0.0	10.67	4.6	9:00a	7:40p	10.67
7–28–36	29	2–36	13.5	8.3	11.33	5.6	7.47	3.1	6.53	2.9	2:28p	9:00p	6.53
8– 3–36	30	2–36	7.2	3.1	7.50	3.8	10.48	4.5	3.52	1.6	5:29p	9:00p	3.52
8– 9–36	31	2–36	13.7	6.8	7.00	3.7	9.72	4.2	3.78	1.7	4:43p	8:30p	3.78
8–10–36	32	2–36	7.5	4.4	9.98	4.7	8.88	3.8	3.45	1.6	3:53p	7:20p	3.45
8–13–36	33	2–36	17.1	9.8	12.00	6.2	6.75	3.0	10.00	4.6	1:45p	11:45p	10.00
8–14–36	34	2–36	10.4	3.1	7.25	3.5	8.67	3.8	17.33	7.6	3:40p	9:00a	17.33
8–15–36	35	2–36	16.1	5.4	0.00	0.0	0.00	0.0	24.00	10.4	9:00a	9:00a	24.00
8–16–36	36	2–36	15.7	8.9	0.00	0.0	0.00	0.0	13.60	6.1	9:00a	10:36p	13.60
8–17–36	37	2–36	9.5	4.7	8.40	4.3	9.43	4.1	5.57	2.3	4:26p	10:00p	5.57
8–18–36	38	2–36	21.8	10.7	9.00	4.7	6.20	2.7	19.80	8.6	1:12p	9:00a	19.80
8–19–30	39	2–36	15.9	12.4	0.00	0.0	0.00	0.0	13.30	5.9	9:00a	10:18a	13.30
8–20–36	40	1–36	15.8	6.7	9.45	4.9	4.98	2.1	20.27	8.9	12:44p	9:00a	20.27
8–21–36	41	1–36	20.1	9.5	0.00	0.0	0.00	0.0	24.00	10.5	9:00a	9:00a	24.00
8–22–36	42	1–36	20.0	9.6	0.00	0.0	0.00	0.0	24.00	10.6	9:00a	9:00a	24.00
8–23–36	43	1–36	17.2	9.6	0.00	0.0	0.00	0.0	14.00	6.1	9:00a	11:00p	14.00
8–24–36	44	1–36	16.9	7.1	8.00	4.0	4.52	2.0	21.48	9.5	11:31a	9:00a	21.48
8–25–36	45	1–36	15.7	10.9	0.00	0.0	0.00	0.0	12.33	5.4	9:00a	9:20p	12.33
8–26–36	46	1–36	18.9	3.6	9.60	4.9	6.85	3.0	19.15	8.3	1:51p	9:00a	19.15
8–27–36	47	1–36	8.7	5.3	0.00	0.0	6.52	2.9	5.48	2.5	3:31p	9:00p	5.48
8–28–36	48	1–36	9.3	2.2	10.00	4.9	9.32	4.1	2.68	1.3	4:19p	7:00p	2.68
9–11–36	49	1–36	14.8	3.5	11.00	5.5	12.50	5.3	4.33	2.0	7:30p	11:50p	4.33
9–14–36	50	1–36	11.3	5.5	11.50	5.6	8.37	3.7	6.13	2.7	3:22p	9:30p	6.13
9–15–36	51	1–36	10.3	4.4	9.50	4.6	9.65	4.1	4.85	2.1	4:39p	9:30p	4.85

TABLE 2 (*Continued*)—COMPLETE DATA AND RESULTS FOR TESTS

DATE 1936	TEST NO.	SERIES NO.	WATER CIRCULATION THROUGH COOLING COIL		WATER CIRCULATION THROUGH COOLING COIL		TOTAL MOISTURE CONDENSED LB	HEAT ABSORBED BY COOLING COIL DURING OVERALL TEST PERIOD, BTU			RATIO MOISTURE TO TOTAL LOAD
			Time in Hours	Quantity in Gallons	Temp. at Inlet F	Temp. Rise Through Coil F		Load Due to Moisture	Sensible	Total	
1	2	3	26	27	28	29	30	31	32	33	34
6–17–36	1	1–36	3.08	778	52.1	9.6	0.00	0	62,300	62,300	0.000
6–20–36	2	1–36	1.95	473	51.6	10.1	0.00	0	39,800	39,800	0.000
6–27–36	3	1–36	4.35	1099	51.2	10.6	4.19	4,400	92,700	97,100	0.045
6–28–36	4	1–36	6.80	1463	52.0	10.5	1.01	1,060	126,940	128,000	0.008
6–29–36	5	1–36	8.25	2082	52.4	10.4	3.53	3,710	176,730	180,440	0.021
7– 2–36	7	1–36	0.83	207	52.2	10.3	0.00	0	17,750	17,750	0.000
7– 3–36	8	1–36	2.42	607	52.5	10.4	0.00	0	52,550	52,550	0.000
7– 4–36	9	1–36	12.55	3155	52.7	10.7	15.77	16,550	265,030	281,580	0.059
7– 5–36	10	1–36	19.53	4950	53.2	10.3	26.82	28,150	396,650	424,800	0.066
7– 6–36	11	1–36	20.18	5025	52.6	11.2	39.64	41,630	427,870	469,500	0.089
7– 7–36	12	1–36	21.48	5457	51.7	11.4	45.49	47,760	470,480	518,240	0.092
7– 8–36	13	1a–36	13.00	3955	52.1	10.0	29.44	30,920	298,530	329,450	0.094
7– 9–36	14	1a–36	17.87	5452	52.1	9.7	50.16	52,670	387,870	440,540	0.120
7–10–36	15	2–36	19.35	3635	47.4	16.0	49.21	51,670	432,800	484,470	0.107
7–11–36	16	2–36	21.33	3460	46.9	18.0	57.40	60,300	459,200	519,500	0.116
7–12–36	17	2–36	20.93	3379	47.0	18.0	49.53	52,000	455,500	507,500	0.103
7–13–36	18	2–36	21.02	3390	46.7	18.5	62.28	65,400	457,600	523,000	0.125
7–14–36	19	2–36	20.80	3397	46.4	18.2	42.44	46,570	470,930	515,500	0.086
7–15–36	20	2–36	16.05	2638	46.7	17.8	62.09	65,200	326,440	391,640	0.167
7–16–36	21	2–36	12.55	2071	46.8	17.7	52.72	55,360	250,050	305,410	0.181

7–17–36	22	2–36	9.70	1584	45.5	17.9	9.55	10,030	226,470	236,500	0.042
7–18–36	23	2–36	2.68	433	49.4	16.2	6.07	6,370	52,120	58,490	0.109
7–23–36	24	2–36	8.17	1338	46.2	18.2	19.92	20,920	182,180	203,100	0.103
7–24–36	25	2–36	1.22	197	46.1	17.3	0.33	350	28,090	28,440	0.012
7–25–36	26	2–36	2.18	358	47.6	19.5	10.60	11,130	47,170	58,300	0.191
7–26–36	27	2–36	13.15	2150	46.4	18.6	56.25	59,050	274,550	333,600	0.177
7–27–36	28	2–36	9.65	1575	46.3	18.7	38.78	40,720	204,780	245,500	0.166
7–28–36	29	2–36	3.95	647	46.3	18.5	9.44	9,910	89,890	99,800	0.099
8– 3–36	30	2–36	1.20	196	45.9	18.1	0.62	650	29,000	29,650	0.022
8– 9–36	31	2–36	1.18	194	46.2	18.0	0.73	770	28,370	29,140	0.026
8–10–36	32	2–36	1.31	215	46.3	18.2	1.54	1,620	31,070	32,690	0.050
8–13–36	33	2–36	7.28	1186	46.1	17.9	26.36	27,690	149,410	177,100	0.156
8–14–36	34	2–36	2.15	348	46.4	19.3	4.02	4,220	51,900	56,120	0.075
8–15–36	35	2–36	9.80	1601	47.0	18.0	47.10	49,440	191,060	240,500	0.206
8–16–36	36	2–36	9.75	1490	46.6	18.4	39.15	41,100	187,800	228,900	0.180
8–17–36	37	2–36	1.85	301	45.8	17.8	1.30	1,370	43,330	44,700	0.031
8–18–36	38	2–36	11.83	1930	46.1	18.1	32.95	34,600	254,800	280,400	0.120
8–19–36	39	2–36	11.17	1818	46.2	18.1	44.69	46,930	227,560	274,490	0.171
8–20–36	40	1–36	7.41	1980	52.2	11.2	23.95	25,150	159,750	184,900	0.136
8–21–36	41	1–36	14.18	3589	52.0	10.9	45.26	47,525	278,745	326,270	0.146
8–22–36	42	1–36	14.75	3732	52.1	11.1	47.74	50,125	295,350	345,475	0.145
8–23–36	43	1–36	9.82	2486	51.8	11.1	33.98	35,680	194,500	230,180	0.155
8–24–36	44	1–36	10.10	2570	52.2	11.3	35.20	36,960	205,245	242,205	0.153
8–25–36	45	1–36	10.33	2621	52.0	11.2	37.52	39,395	205,380	244,775	0.161
8–26–36	46	1–36	7.05	1796	51.9	11.1	32.11	33,715	132,520	166,235	0.203
8–27–36	47	1–36	1.60	410	52.2	11.4	1.56	1,640	37,290	38,930	0.042
8–28–36	48	1–36	0.95	243	52.4	11.9	0.96	1,010	23,050	24,060	0.042
9–11–36	49	1–36	0.90	229	....	....	0.24	250	20,340	20,590	0.012
9–14–36	50	1–36	1.50	382	52.0	11.9	0.94	990	36,890	37,880	0.026
9–15–36	51	1–36	0.98	250	52.2	11.7	0.30	315	24,060	24,375	0.013

From the top set of curves in Fig. 4 it may be observed that on this day, in common with a number of other similar days occurring during the summer of 1936, the effective temperature of the outdoor air was always greater than the effective temperature of the indoor air. Hence no cooling could be effected during the night by the introduction of large quantities of outdoor air. However, this limitation during the extremely hot portion of the summer did not preclude the effective use of outdoor air at night during the greater portion of the season when the maximum outdoor temperature did not exceed approximately 100 F.

Actual Cooling Loads

During the periods of intermittent operation the actual cooling load curve showed high peaks at the start of each operating period. These peaks correlated with periods of high indoor relative humidity, indicating that periods of high wet-bulb temperature for the air entering the coil were accompanied by corresponding periods of comparatively high heat transmission for the coil. Similar conditions were observed in the tests made in 1935 with 58 F water supplied to the cooling coil. It was shown that the heat absorbed by the cooling coil was practically a linear function of the enthalpy of the entering air. Furthermore, periods of high indoor relative humidity also represented periods of high wet-bulb temperature with corresponding high enthalpy for the air entering the coil. Since high heat transmission accompanied high enthalpy for the air entering the coil, it serves to explain the fact that the peaks in the load curve in Fig. 4 occurred at times corresponding to high indoor relative humidity.

In this connection it is of interest to compare the typical results which were obtained from the tests made in 1935 and 1936. In 1935 the indoor dry-bulb temperature was maintained at approximately 80 F for a day in which the maximum outdoor temperature was 96 F, giving a temperature difference of 16 F. The heat absorbed by the 58 F water passing through the cooling coil was 23,700 Btu per hour. In 1936 the indoor dry-bulb temperature was maintained at approximately 80 F on a day in which the maximum outdoor temperature was 104 F, giving a temperature difference of 24 F, and the heat absorbed by the 52 F water passing through the cooling coil was 24,150 Btu per hour. That is, for an increase in temperature difference between the maximum outdoor temperature and the average indoor temperature of 8 F or 50 per cent, the increase in the rate of heat absorption by the cooling coil was only 2 per cent.

This may be accounted for by the fact that in 1935 the relative humidity of the air entering the cooling coil was 62.8 per cent, whereas in 1936 the value was 48 per cent. If the air entering the cooling coil had contained more moisture in 1936, with a correspondingly higher wet-bulb temperature and enthalpy, then since the heat transmission is practically a linear function of the enthalpy of the entering air, a much greater rate of heat transmission would have been obtained. This would probably have been sufficient to compensate for the resulting increase in the latent heat load. The fact that the absorption of sensible heat was only 19 per cent greater in 1936 than it was in 1935 while the increase in sensible heat load was 50 per cent makes it probable that the plant was overloaded in 1936. Further evidence on this

point is given by the fact that, as shown in Fig. 4, the indoor temperature was increasing at 2:00 p.m., and continued to do so up until 6:00 p.m. The total rise, however, was only approximately 1.5 F and had no appreciable influence on the comfort conditions maintained. Thus the heat lag of the structure tended to offset the influence of the higher outdoor temperature occurring in 1936, by preventing an abnormal rise in indoor temperature even though the plant was not able to completely carry the load.

Calculated Cooling Load

The total cooling load imposed on a cooling plant consists of the sum of all heat gains by the structure resulting from indoor-outdoor temperature differences and sun effects on walls and windows, and of all other heat quantities introduced by air inleakage and heat given up by the occupants and by mechanical or electrical processes. These heat quantities may be itemized as follows:

ITEM NO.	HEAT SOURCE
1	Heat transmission through walls not exposed to sun
2	Heat transmission through walls exposed to sun
3	Heat transmission through windows and doors not exposed to sun
4	Heat transmission through windows exposed to sun
5	Heat transmission through ceilings
6	Heat transmission through floors
7	Heat brought in by infiltration or ventilation air
8	Heat given off by occupants
9	Heat given off by electric motors
10	Heat given off by electric lights

Complete details of the method of calculation of the cooling load have been outlined in a previous publication[5] and will not be repeated in this paper. The values of coefficients of heat transmission and enthalpies of mixtures of water vapor and air as given in THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS GUIDE, 1936, have been used.

The most common method of calculating the cooling load, which changes from hour to hour depending on variations in outdoor temperature and sun effect, is based on the assumption that the changes in load occur coincident with the changes in outdoor temperature and sun effect.

If, for example, the temperature difference between the outdoors and indoors is 20 F at a given time, the heat transmission through the wall at that time is calculated on the basis that conditions of thermal equilibrium have been maintained with this 20 F difference and the existing sun effect. However, since the walls offer some resistance to heat flow, changes in outdoor temperature and sun effect are not reflected instantaneously in changes in heat flow, and an appreciable interval exists between the time that a given outdoor temperature and sun effect is imposed and the time at which the heat reaches the indoors. This becomes evident from the fact that the temperature of the inside surface of the wall reaches a maximum considerably later than that attained by the outdoor temperature.

A second method for calculating the cooling load takes into account this time lag in the transmission of heat through the walls and ceiling. In this case the calculated value of the heat transmission through the wall at any

[5] University of Illinois, Engineering Experiment Station, Bulletin No. 290, 1937, A. P. Kratz, M. K. Fahnestock, and S. Konzo.

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given hour is based, not on the temperature difference between the outdoors and indoors and the sun effect occurring at the given time, but on the temperature difference and sun effect that existed at some previous time. Obviously, the lag that exists between the time that a temperature differential is imposed

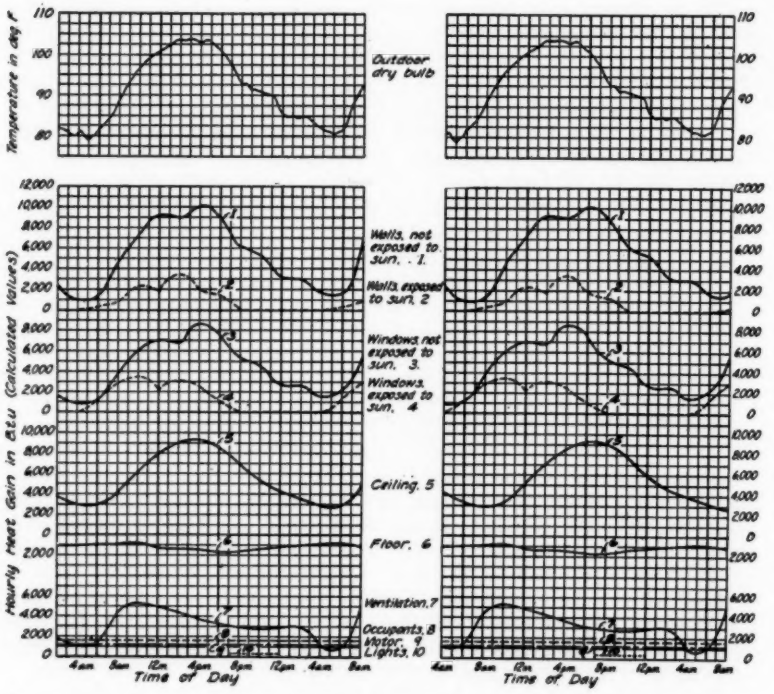

FIG. 5a. CALCULATED COOLING LOAD, NO TIME LAG ASSUMED. CALCULATED COMPONENTS OF COOLING LOAD ON RESIDENCE, JULY 7, 1936, TEST NO. 12, SERIES 1-36, MAXIMUM OUTDOOR TEMPERATURE 104 F

FIG. 5b. CALCULATED COOLING LOAD, TIME LAG ASSUMED. CALCULATED COMPONENTS OF COOLING LOAD ON RESIDENCE, JULY, 7, 1936, TEST NO. 12, SERIES 1-36, MAXIMUM OUTDOOR TEMPERATURE 104 F

on a wall and the time that the inside surface temperature is affected is dependent on the type of construction, the mass, and the conductivity of the wall. A massive wall that is well insulated, for example, will show a greater time lag than a lighter wall that is not as well insulated.

In order to determine the difference in the results obtained with the two methods of calculating the cooling load, the hourly values for the cooling load were first determined by assuming no time lag. These hourly values of the heat transmission were calculated for July 7, 1936. The values of the ten individual items of heat gain are represented in Fig. 5a and the total calculated cooling load obtained by summing the values of the individual items at any

given time is shown in Fig. 4. It may be noted from Fig. 5a that the maximum value for the heat transmission through the walls which were not exposed to the sun (Curve No. 1) was obtained at approximately 4:30 p.m. Similarly the maximum values for the items represented by curves numbered 2 and 5, respectively, were obtained at 2 p.m. and 3:30 p.m.

For the second method of calculating the cooling load the amount of the time lag to be used was determined from experimental data obtained by means of thermo-couples placed on the inside surfaces of the walls and ceiling. These studies indicated that for the conditions maintained in the Research Residence an interval of approximately 2 hours was required for heat to flow through the sidewalls and an interval of approximately 3 hours for heat to flow through the second story ceiling. Hence the heat gains represented by curves Nos. 1 and 2 in Fig. 5a were not changed in magnitude, but were replotted against a new time basis in Fig 5b so that the maximum values were obtained 2 hours later. Similarly the curve representing the heat transmission through the ceiling, curve No. 5a, was replotted in Fig. 5b so that the maximum value was obtained 3 hours later than that shown in Fig. 5a. It may be noted from Fig. 5b that the maximum values for the items represented by curves Nos. 1, 2, and 5 were therefore obtained at 6:30 p.m., 4 p.m., and 6:30 p.m., respectively. The seven remaining items of heat gain were not influenced by heat lag and hence the curves representing these items in Fig. 5b were not changed as compared with the corresponding curves shown in Fig. 5a. The total calculated cooling load was again obtained by summing the individual items in Fig. 5b at any given time and the resulting curve is shown in Fig. 4.

It may be observed from the two curves shown in Fig. 4 for the calculated cooling load on the Residence that the maximum values obtained by the two methods of calculation were practically the same, although the times at which the maximum values were obtained were approximately 1 hour apart. Although the maximum values were obtained at approximately 3 p.m. when no time lag was assumed and at approximately 4 p.m. when a time lag was assumed, the actual temperature of the indoor dry bulb did not attain a maximum value until approximately 6 p.m. This would tend to indicate that in order to obtain a closer correspondence between the calculated cooling load and the actual cooling load, an even greater time lag than that which was used would have to be assumed. However, the use of a greater time lag in the case of the Research Residence could not be justified from the observations of the inside surface temperatures of the walls and ceilings.

In the calculation of the design load the maximum value of the load is of more consequence than is the time at which the maximum occurs. The fact that practically the same maximum value was obtained by both methods of calculation seems to indicate that in the case of frame structures similar to the Research Residence the more complicated method of calculation based on the assumption of time lag is hardly justified, although it may be justified for more massive structures.

The curve based on the assumption of the existence of a time lag showed, in general, a more gradual rate of increase of the cooling load until the maximum value was reached and also a less rapid decrease of the cooling load after the maximum value was attained than was shown by the curve based on the assumption of the existence of no time lag. It should be noted

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that the actual heat absorption during this entire period remained constant in spite of the fact that the indoor dry-bulb temperature slightly exceeded the predetermined value of 80 F. Hence, if the capacity of the cooling plant had not been limited and if the actual rate of heat absorption by the coil could have been increased by increasing the rate of water flow so that the indoor temperature had been maintained at a constant value of 80 F during the period from 2 p.m. to 10 p.m., it seems likely that the curve representing the actual heat absorption would have attained a closer correspondence to the calculated curve based on the assumption of time lag than to the curve based on the assumption of no time lag. Under these conditions, it is possible that the method of calculation based on the assumption of a time lag may have some merit and consideration of it should not be entirely neglected in design procedure,[6] particularly in the case of massive structures.

TABLE 3—MAXIMUM CALCULATED VALUES FOR THE TEN COMPONENT ITEMS OF THE COOLING LOAD

Based on Test No. 12, July 7, 1936. Research Residence

ITEM NO.	SOURCE OF HEAT GAIN	MAXIMUM VALUE BTU PER HOUR	PER CENT OF TOTAL OF MAXIMUM VALUES
1	Walls, not exposed to sun	10,100	24.2
2	Walls, exposed to sun	3,400	8.2
3	Windows, not exposed to sun	8,700	20.9
4	Windows, exposed to sun	3,500	8.4
5	Ceiling	9,300	22.3
6	Floor	−1,700	−4.1
7	Ventilation	5,300	12.7
8	Occupants	1,600	3.8
9	Motor	1,200	2.9
10	Lights	300	0.7
	Total[a] of maximum values	41,700	100.0

[a] The total of the maximum values of the component items is approximately 13 per cent greater than the calculated value of the maximum cooling load that was obtained at 4 p.m.

Possible Reductions in Cooling Load

The curves shown in Fig. 5b are of additional interest in that they present in a graphical manner the relative magnitudes of the ten component items that comprised the total calculated cooling load on the Residence on a typical day, in which the outdoor temperature attained a maximum value of 104 F. The maximum values attained by each component item, irrespective of the times at which they occurred have been listed in Table 3.

For this particular case, practically one-quarter of the total value may be attributed to the heat transmission through the walls which were not exposed to the sun and approximately one-twelfth to the heat transmission through walls which were exposed to the sun. That is, approximately one-third of the total may be attributed to the heat transmission through the walls alone. It

[6] A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads, F. H. Faust, L. Levine, and F. O. Urban, A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 327.

may be observed that by the application of effective insulation to the uninsulated side walls, for which the coefficient of heat transmission was 0.25 Btu per square foot per hour per degree difference in temperature, substantial reductions in the heat gain would be possible. This reduction in the heat gain through the walls themselves would be proportional to the reduction in the coefficient of heat transmission. The percentage reduction in the total heat gain would be one-third of that through the walls alone. In addition, the time lag for the passage of heat through the walls would also be increased so that the cooling load would be distributed more uniformly with respect to the time of day. In a similar manner substantial reductions would be possible in the heat transmission through the windows which were not exposed to the sun, provided double-glass windows could be used in place of the single-glass windows. Since all of the windows exposed to the sun were equipped with awnings, the value listed as item 4 was considerably smaller than that which would have been obtained had no awnings been used.

It may be observed from Table 3 that the maximum heat gain through the walls exposed to the sun was 3400 Btu per hour, or only 8.2 per cent of the sum of all maximum heat gains. The indoor-outdoor temperature difference occurring at this time was 23.4 F, and the total area exposed to the sun was 463 sq ft. If the sun effect at this time had been neglected, and the heat transfer calculated as merely that resulting from the temperature difference, a value of 2720 Btu per hour would have been obtained. Comparing this with the 3400 Btu per hour obtained by computations involving the sun effect, it is evident that the difference is only 680 Btu per hour, or 1.6 per cent of the sum of all maximum heat gains. Since this is within the limits of accuracy of any of the methods used for computing heat gains, it would seem that, for the purpose of obtaining the maximum load to be used for design, the more involved method taking into account the sun effect on the walls can hardly be justified in the case of a structure similar to the Research Residence. This is not true of windows exposed to the sun, however, because if the windows had not been shaded by awnings, the resulting heat gain would have been approximately 12,100 Btu per hour instead of the 3500 shown in Table 3, and would have constituted a large portion of the total heat gain. Hence the sun effect on windows, and particularly on those which are unshaded, cannot be neglected.

The amount of heat transmitted through the ceiling (Item 5) is exceedingly large, due not only to the relatively high coefficient of heat transmission, but also to the very large temperature difference that exists between the attic space and the indoor air. The effective use of insulation should serve to materially reduce the heat gain from this source also. The heat brought in by ventilation and infiltration (Item 7) is dependent not only on the amount of air introduced into the house but also on the climatic conditions that exist. In this connection, it may be noted that when the moisture content of the outdoor air is high, as was the case in 1935,[7] the numerical value of Item 7 may be much greater than that listed in Table 3. Restricting the amount of air required for ventilation tends to accentuate any odors, and further reduction in the number of air changes may not be advisable in the average residence. The values listed as Items 8, 9, and 10 are usually insignificant in the case of resi-

[7] The load due to ventilation at 2 p.m. on August 2, 1935, was 8715 Btu per hour.

dence installations, although they may be of considerable importance in commercial or industrial installations. An analysis of the components of the calculated cooling load therefore indicates that the size of the cooling unit required and the cost of operation of the unit may be materially reduced by substantial improvements in the construction of the house. The use of awnings on all windows exposed to the sun, of double-glass in windows, and of effective insulation both in the sidewalls and in second story ceilings or attic floors should be encouraged if summer cooling of residences is to become a practical matter.

Humidity Conditions Maintained

The curves in Fig. 6 show that during periods of continuous operation, both in the case of the water coil used in the studies made in 1935 and 1936 and in the case of the mechanical refrigerating unit used in the study of 1934, the

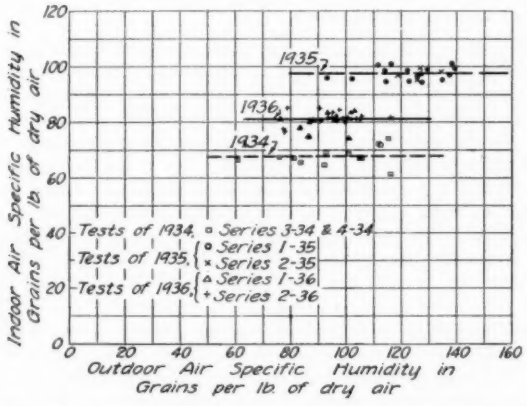
Indoor Air Specific Humidity in Grains per lb. of dry air
120
100
80
60
40
20
0
1935
1936
1934
Tests of 1934, ○ Series 3-34 & 4-34
Tests of 1935, ○ Series 1-35, × Series 2-35
Tests of 1936, △ Series 1-36, + Series 2-36
0 20 40 60 80 100 120 140 160
Outdoor Air Specific Humidity in Grains per lb. of dry air

FIG. 6. SPECIFIC HUMIDITY OF INDOOR AIR RESULTING FROM OPERATION WITH TWO TYPES OF COOLING SYSTEMS

indoor specific humidity maintained was dependent only on the temperature of the medium used and the character of the cooling coil and was independent of the amount of water vapor in the outdoor air. Since in both cases a constant indoor dry-bulb temperature of 80 F was maintained, the specific humidity is also representative of the indoor relative humidity. That is, the water coil used in 1935 with a mean water temperature of approximately 62 F maintained a constant indoor relative humidity of 63 per cent during periods of continuous operation, while the water coil used in 1936 with a mean water temperature of 57.6 F and 55.5 F for Series 1-36 and Series 2-36 respectively, maintained a constant relative humidity of approximately 52 per cent for both cases. The evaporator coil used in 1934 with a mean temperature of approximately 45 F for the refrigerant maintained a constant relative humidity of 45 per cent.

Possibly by coincidence the numerical value of the relative humidity that was maintained in the house was nearly the same as the numerical value of the mean coil temperature. Ordinarily for small cooling installations, in which the selection of the size of the cooling coil is governed by considerations based

only on the maintenance of some given value of the indoor dry-bulb temperature, the indoor relative humidity accompanying this temperature is more or less indeterminate. Unless special efforts are made either to dehumidify the air by the use of additional equipment, or to reheat the air either directly or otherwise, the indoor relative humidity that will be obtained cannot be maintained at any arbitrarily selected value. Over a limited range of mean coil temperatures and under test conditions similar to those maintained in the Research Residence, the suggested approximation may be of some value in determining the probable magnitude of the indoor relative humidity in similar residence installations. On the assumption that this relationship is sufficiently general until proved otherwise, it might serve as an index for the selection of desirable coil temperatures.

During periods of operation the ventilating air from outdoors was mixed

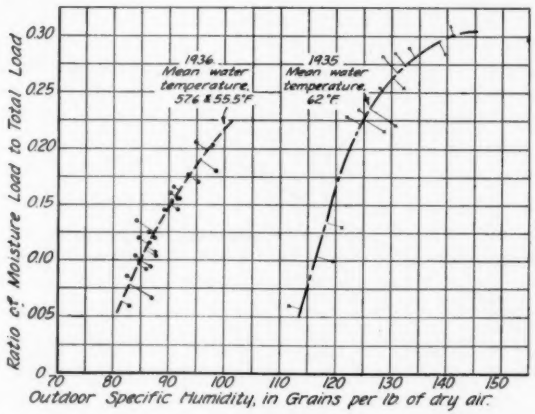

FIG. 7. RATIO OF THE LOAD DUE TO CONDENSED MOISTURE TO THE TOTAL COOLING LOAD

with the recirculated air from the house just before entering the cooling coil. The moisture content of the outdoor air therefore directly influenced that of the mixed air entering the coil. In the tests reported in 1935 it was indicated that the enthalpy of the entering air increased as the specific humidity increased, resulting in an increase in the heat transmission for the coil. At the same time the specific humidity of the air leaving the coil increased, but not to as great an extent as that of the inlet air. This indicated that a part, if not all, of the increase in heat transmission was utilized in increasing the amount of condensation as the specific humidity of the outdoor air and hence that of the air entering the coil, was increased. Thus the increasing heat transmission of the coil tended to compensate for increases in the outdoor specific humidity, and, as a result, also tended to maintain a constant indoor relative humidity during periods of continuous operation, as shown in Fig. 6.

The maintenance of a practically constant indoor relative humidity accompanying rather wide variation in the outdoor humidity conditions would imply that the proportion of the total cooling load that could be attributed to the

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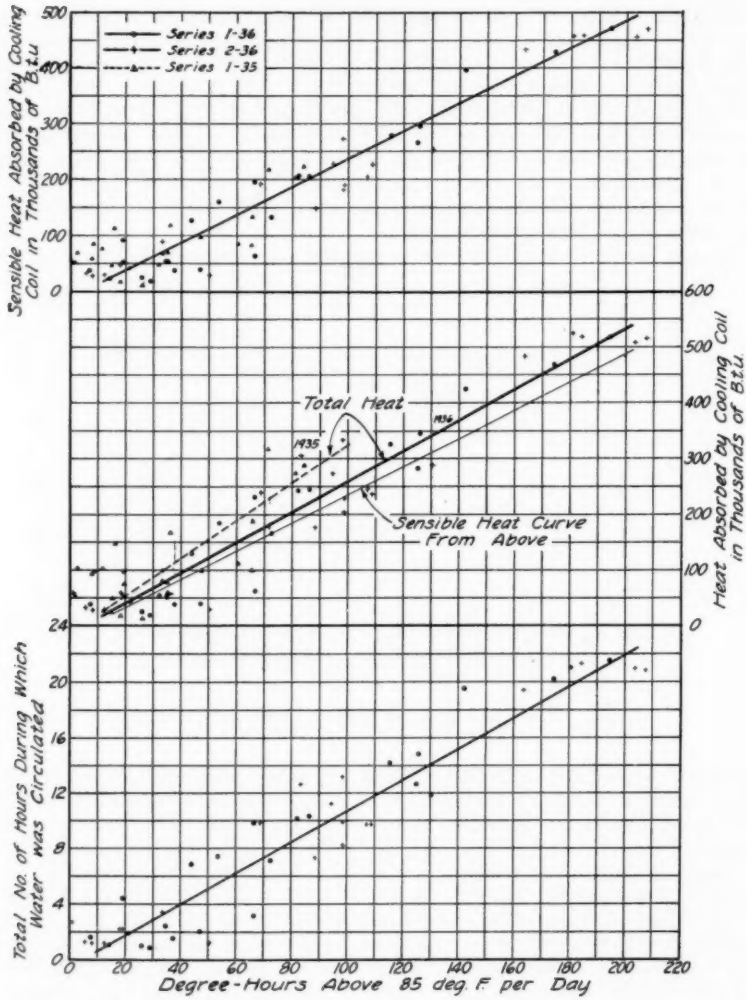

FIG. 8. HEAT ABSORBED BY COOLING COILS AND HOURS OF WATER CIRCULATION PER DAY

process of dehumidification would increase as the outdoor humidity increased. The curves presented in Fig. 7 indicate that such was the case. It may be observed from the results obtained in the 1936 tests that the ratio of the moisture load to the total load increased from approximately 0.05 to 0.22 when the outdoor specific humidity increased from 80 grains to 100 grains per pound of air. A similar increase may be observed for the tests made in 1935, although both the lowest value and the total range of the outdoor specific humidity were definitely greater in that year, indicating a much wetter season than the one for 1936. From extrapolation of the curve for 1936 in Fig. 7 it is evident that had the 1936 season been characterized by as much moisture in the outdoor air as was the case for the 1935 season, the ratios would have been equal to or greater than those obtained in the 1935 tests. Therefore, it is apparent that the relative magnitude of the load resulting from the moisture condensed was dependent not only on the mean coil temperature used but also on the amount of moisture contained in the outdoor air. Furthermore, since the extrapolated portion of the curve for 1936 would lie above the curve for 1935, it seems probable that the increased heat transfer, resulting from the lower mean coil temperature and the increased enthalpy of the entering air, would have made it possible to have maintained a lower indoor relative humidity in 1936 than was maintained in 1935 even if the 1936 season had been as damp as the one for 1935.

Heat Absorbed per Day

The total cooling load for the day, expressed as the total Btu absorbed from the air in the total amount of time during which water was actually being circulated is shown in Fig. 8 plotted against the degree-hours above 85 F per day. The base temperature of 85 F was selected because it was found that usually no cooling was required unless the outdoor temperature was 85 F or more. Curves for both total heat and sensible heat are shown, and the difference between these curves represents the heat due to the change in the moisture content of the air. Considerable deviations from the curves representing median conditions occurred in the individual points, each of which was obtained from data based on a single day's operation. These deviations could be accounted for by variations in the wind and sun effects which influence the load but are not directly dependent upon the degree-hours. In addition, if the night was exceptionally cool, less than the average amount of artificial cooling was required the next day, while if the night was exceptionally warm the reverse was true.

From the upper group of curves in Fig. 8 it may be noted that the sensible heat absorbed during the tests made in 1935 and during the two series of tests made in 1936 could be represented by a single curve. This would indicate that the amount of sensible heat absorbed by the cooling coil for a given outdoor condition is independent of the coil temperatures and is dependent only on the temperature difference maintained between the outdoor air and the indoor air. On the other hand the middle group of curves in Fig. 8 indicates that the total heat absorbed by the cooling coil was greater for the tests made during the 1935 season than for the tests made during 1936. This condition was caused by the fact that a relatively greater amount of heat was required to reduce the moisture content of the air during the former season, which was characterized by a high moisture content in the outdoor air. The

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curves in the lower portion of Fig. 8 show the total number of hours per day during which water was circulated. For those days on which the outdoor temperature attained a maximum value between 104 F and 108 F the

TABLE 4—SUMMARY OF RESULTS OF TESTS USING WATER FOR COOLING AND USING NIGHT AIR COOLING FOR ENTIRE SEASON OF 1936

	WEATHER DATA		
1	Total hours above 85 F for season of 1936		706.3
2	Total hours above 90 F for season of 1936		341.6
3	Total degree-hours above 85 F for season of 1936		4551.0
4	Total degree-hours above 90 F for season of 1936		1962.8
	NIGHT AIR COOLING		
5	Total number of nights with night air cooling by fan		62
6	Total running time for fan during night air cooling, hours		614.4
7	Average rate of power input to fan during night air cooling, watts		503
8	Total power input to fan during night air cooling, kilowatt hour		308.9
	AIR RECIRCULATION WITHOUT WATER COOLING		
9	Total number of days with recirculation of house air by fan		63
10	Total running time for fan during recirculation of house air, hours		621.0
11	Average rate of power input to fan during recirculation without water cooling, watts		441
12	Total power input to fan during recirculation of house air without cooling, kilowatt hour		273.5
	AIR RECIRCULATION WITH WATER COOLING		
13	Total number of tests with water cooling		51
14	Total running time for fan during test period, hours		695.0
15	Average rate of power input to fan during test period, watts		442
16	Total power input to fan during test period, kilowatt hours		307.5
	COMBINED TOTALS		
17	Total running time for fan including night air cooling, hours		1930.4
18	Total power input to fan including night air cooling, kilowatt hours		887.7
19	Total number of hours of test period for season		695.0
20	Total number of hours of water circulation during test period.		452.1
21	Quantity of cooling water		
	Gallons, Series 1	54,991	
	Series 2	39,531	
	Total		94,522
	Cu ft, Series 1	7329	
	Series 2	5271	
	Total		12,600

operation of the plant was practically continuous, since the cold water was circulated approximately 22 hours out of the 24. These curves shown in Fig. 8 can be used to estimate the seasonal amount of heat absorption and the total time of plant operation for any season for which the number of degree-hours above 85 F corresponding to each day in the season is known.

A comparison of the sensible heat absorbed by the coil supplied with cold water in 1935 and 1936 with that absorbed by the expansion coil used in connection with the mechanical refrigerating unit employed in 1934 affords a means of determining the magnitude of the heat gain introduced into the house by the presence of the refrigerating machine in the basement. This is true because the heat emitted from the machine would appear as a sensible

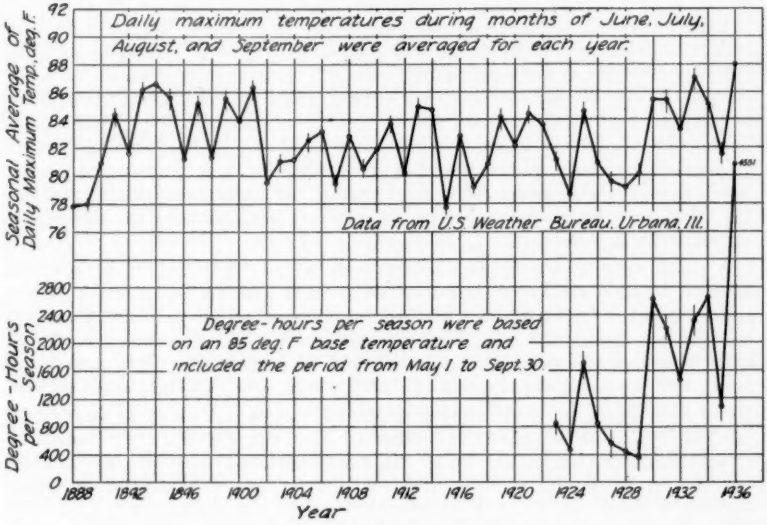

FIG. 9. GRAPHICAL RECORD OF SUMMER TEMPERATURES IN URBANA, ILL., FROM 1888 TO 1936

heat gain alone, and therefore the difference in latent heat loads for the two seasons may be disregarded. In the case of the coil supplied with cold water the amount of sensible heat removed from the air by the water is equal to the net sensible heat gain of the house. However, in the case of the mechanical refrigerating unit located in the basement, the sensible heat gains contributed by the unit, such as the electrical and mechanical losses in the compressor motor, the mechanical losses in the compressor and drive, and the heat losses from the condenser and compressor heads constituted an additional load in excess of the net sensible heat gain of the house. The result was such that on two similar days characterized by the same number of degree-hours the sensible heat absorbed by the direct expansion coil used in connection with the mechanical unit was greater than that absorbed by the water coil. It was observed that for a day having 80-degree-hours, the sensible heat absorbed by the mechanical unit was 250,000 Btu [8] per day and by the water cooling coil was 190,000 Btu per day. The difference of 60,000 Btu per day amounted to 24

[8] A.S.H.V.E. Research Paper, Study of Summer Cooling in the Research Residence for the Summer of 1934, A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick, A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 207.

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per cent of the heat absorbed by the mechanical unit. The ratio of 190,000 to 250,000 may be regarded as an approximate index of the overall house efficiency of the mechanical unit. This ratio of 0.76 agrees remarkably well with the value of 0.767 obtained directly from an analysis of the heat balance on the house, as presented in a previous paper.[9]

Summary of Seasonal Results

Table 4 gives a summary of the total quantities obtained for the season of 1936. This season was the hottest summer occurring in Urbana, Illinois, since the year 1888, when the local Station of the U. S. Weather Bureau was established. As may be noted from Fig. 9 the average of the 122 daily maximum temperatures that were obtained during the months of June, July, August, and September was 88.0 F, which was approximately 1 deg higher than the greatest average of the daily maximums obtained during the previous 49 years. The total of 4551 degree-hours above 85 F, which was almost twice that obtained for 1934, is a better indication of the extreme weather conditions that prevailed during the summer.

A comparison of the weather conditions between 1936 and 1935 is of special interest, since in the latter year tests were made with 58 F water supplied to the cooling coil. The number of days with temperatures above 85 F was 81 for the season of 1936 and 55 for the season of 1935. The number of days with temperatures above 90 F were 50 and 19 respectively. The number of hours with temperatures above 85 F for the season of 1936 was 706.3 as compared with 322.6 for 1935, and the number of hours with temperatures exceeding 90 F were 341.6 and 82.5 respectively. The severity of the season as measured by degree-hours is apparently determined largely by the total number of days on which the maximum temperature exceeds 90 F, since the higher number of degree-hours is always accompanied by the higher number of hours with temperatures above 90 F. While the 1936 season was severe from the standpoint of outdoor temperatures, it was characterized by much lower outdoor humidities than were obtained for 1935. This is indicated to a certain extent by Fig. 7 from which it may be observed that the outdoor specific humidity ranged from 80 to 100 grains per pound of dry air in 1936 as compared with a range of from 110 to 140 grains per pound for 1935.

From Item 6 in Table 4 it may be observed that the total time that the fan was running during periods when it was circulating air from outdoors at night was 614.4 hours. At the local rate for electrical current of 3.1 cents per kilowatt hour prevailing in Urbana, Ill., the cost of operating the fan during these periods was 1.56 cents per hour. During the stand-by periods from 7 a.m. until the indoor temperature rose to 80 F the fan was allowed to recirculate the air in the house. As shown by Item 10 the total running time for the fan during these periods was 621.0 hours. During all of the time that cooling was demanded, the fan was allowed to run irrespective of whether the water circulating pump was operated or not. The total time that the fan was running during these periods, as shown by Item 14, was 695.0 hours. The total time during which the fan was operated to recirculate the air in the house was therefore 1316.0 hours, and the cost of operating the fan to recirculate the air in the house was 1.36 cents per hour.

[9] Loc. cit. See Note 8.

The total amount of water actually used for tests made under conditions of Series 1-36 during the total running time of 211.9 hours was 54,991 gal, and for tests made under conditions of Series 2-36 during the total running time of 240.2 hours was 39,531 gal. At the prevailing rate of 33 cents per thousand gallons, which was also the same as the prevailing rate for the tests of 1935, the hourly costs of water were 8.56 cents and 5.43 cents for Series 1-36 and 2.36 respectively. The total hourly cost for both water and electricity during the actual time when the water was running was 9.9 cents for Series 1-36 and 6.8 cents for Series 2-36, as compared with 13.1 cents for the tests made in 1935. It may be noted that the use of colder water for the cooling coil permitted a marked reduction in the required amount of cooling water, and for equal unit costs for the water, this was reflected in a substantial reduction in the operating costs.

Owing to wide variations in seasonal demands and in local water and electrical rates, seasonal costs are comparatively meaningless. With restriction of due appreciation of their limitations, however, they are undoubtedly of some interest. The first part of the summer from May 1 to June 17 was quite cool and the house temperatures did not attain 81 F at any time during this entire period. Therefore, the actual cooling season extended from June 17, 1936 to September 16, 1936. The total cost for electricity for operating the fan both to circulate air from outdoors at night and to recirculate the air in the house during the day was $27.52. The total cost for water was $31.19 and the cost for both electricity and water was $58.71.

In considering the possible application of cold water as a medium for cooling structures similar to the Research Residence some attention should be given to both the advantages and the disadvantages of the method. These may be itemized as follows:

a. DISADVANTAGES OF SYSTEM

1. The method is confined to those localities in which an abundant supply of cold water is available.

Since water temperatures exceeding 60 F are not suitable, the method is not applicable to those localities supplied with relatively warm surface water. Furthermore, even for localities having a cold water supply, limitations may be imposed as to the quantity of cooling water available for any given installation.

2. High mean coil temperatures are accompanied by high indoor relative humidities. However, the humidity values accompanying mean coil temperatures of approximately 56 F are acceptable.

3. A relatively large amount of coil surface is required. In order to minimize the amount of cooling coil surface correspondingly greater amounts of cooling water must be used.

4. On account of the resistance imposed by the cooling coil to the flow of air, a fan must be selected, which is capable of delivering air against a comparatively high resistance. The use of larger face areas of the cooling coil and the use of lesser number of sections will reduce this air resistance.

5. If the cooling water is supplied from city water mains, the large quantity of water used may materially reduce the water pressure in the service lines to the building. In some cases this may necessitate the use of a larger service line.

b. ADVANTAGES OF SYSTEM

1. The system is characterized by simplicity, both of installation and of operation. The controls required for the system are also simple in type and operation.

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TABLE 5—TEMPERATURE DISTRIBUTION IN FOUR CORNERS OF ROOM

ARRANGEMENT	DESCRIPTION	DEVIATIONS IN TEMPERATURE FROM THOSE OBTAINED AT CENTER OF ROOM[a]									
		BREATHING LEVEL					FLOOR				
		NW	NE	SW	SE	Aver.	NW	NE	SW	SE	Avg
A	Baseboard, Louvers horizontal...	+0.2	−0.5	+0.1	+0.2	0.0	−4.1[b]	−3.1[b]	+1.1	+0.9	−1.3
B	Baseboard, Louvers 60 deg.....	−2.0[b]	−0.2	+0.6	+0.6	−0.2	−0.7	−0.2	+0.2	+0.3	0.0
C	Sidewall, Louvers horizontal...	−0.4	−1.6[b]	0.0	+0.1	−0.5	+0.4	−0.4	+0.5	+0.6	+0.3

[a] Values preceded by (+) were for temperatures greater than, and values preceded by (−) were for temperatures less than those obtained at the center of the room.
[b] These values were affected to some extent by the cool air from the register.

2. The number of moving parts is few and little servicing is required.
3. The operating costs are reasonably low.
4. The unit is extremely quiet in operation.
5. The water used for cooling may be used for sprinkling the lawn.
6. No additional heat from the operation of the cooling unit is added to the heat load of the house.

Register Tests

Three different arrangements of the register were investigated, as illustrated by the insets in the upper portion of Fig. 10. In arrangement *A* the register was located in the baseboard and the louvers were adjusted so that the cool air was projected horizontally into the room near the floor. For arrangement *B* the louvers were set at an angle of 60 deg with the horizontal so that the air was projected upwards into the room. For arrangement *C* the register was located in the sidewall at a height of 7 ft from the floor. In this case the air was delivered horizontally into the upper part of the room.

In Fig. 10 curves are shown giving the relation between the outdoor-indoor temperature difference and the difference in temperatures between the 5 ft level and the floor, and between the ceiling and the 5 ft level. These room air temperatures were obtained by means of thermometers located on the central axis of the room. However, as shown by Table 5, other temperatures observed by means of thermometers located in the four corners of the room indicated that at the 5 ft level the average value of the deviations of these temperatures from those observed in the center of the room was not greater than 0.5 F. It may be noted from Table 5 that some of the temperatures obtained in the north half of the room were affected to some extent by the cool air from the registers. In general, the temperature distribution in all parts of the room was quite uniform, and hence the values obtained in the center may be regarded as typical of those obtained in all parts of the room.

It may be observed from Fig. 10 that in all cases the temperature gradient from the floor to the ceiling tended to increase as the outdoor temperature increased. For an outdoor temperature of approximately 100 F, or for an outdoor-indoor difference of 20 F, the total differences in temperature from the floor to the ceiling were 6.2 F, 2.5 F, and 3.2 F for arrangements *A*, *B*, and *C* respectively. The temperature gradients obtained for arrangements *B* and *C* were quite small, and even that for arrangement *A* was not excessively

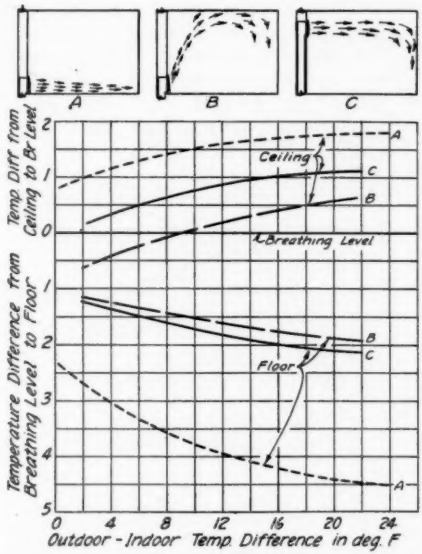

FIG. 10. ROOM AIR TEMPERATURES WITH THREE TYPES OF REGISTERS

large. Furthermore, temperatures in the living zone lower than those in the upper part of the room are not as undesirable in the summer time as they are in the winter. Therefore, it is evident that the performance of registers used for summer cooling work should be evaluated from some standpoint other than that of the temperature gradients produced in the room, and the absence of objectionable air currents or drafts in the living zone is suggested as a better criterion for judging the performance of registers in this case. As indicated by the diagram in Fig. 11, the use of arrangement *A* produced appreciable air currents directly in front of the register, and proved to be unsatisfactory from this standpoint. The air was projected across the floor to the opposite wall at a rather high velocity. This is indicated by the numerical values, for the air velocity in feet per minute, shown in Fig. 11. These values are placed adjacent to the arrows indicating the direction and magnitude of the air flow, and owing to limitations of the instruments used for observing very low velocities, the lower values shown must be regarded as only approximate. The path of the main air stream was such that a considerable portion

of the lower part of the room was subjected to objectionable drafts. Although the use of a diffusing type of register, which would tend to distribute the air outwards into the room, might serve to reduce the magnitude of the air velocity, it is probable that objectionable drafts would still be obtained in a

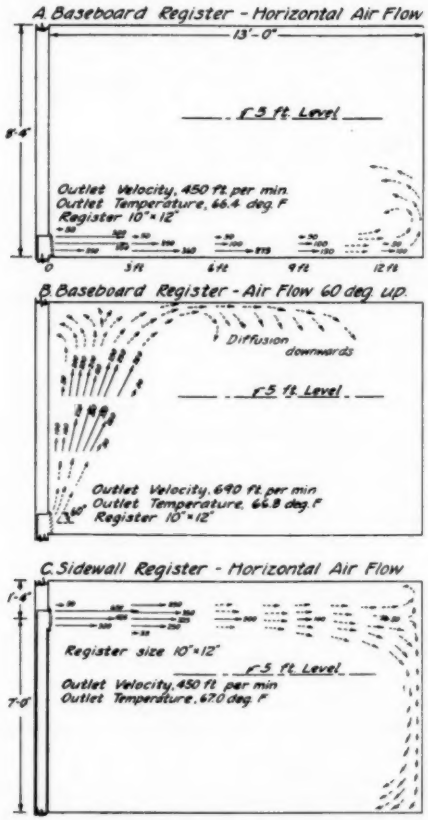

FIG. 11. DIAGRAMS FOR AIR CIRCULATION IN ROOM WITH THREE TYPES OF REGISTER AIR OUTLETS

considerable zone located in front of the register. Therefore, it may be concluded that a baseboard register with horizontal louvers of the type indicated by arrangement *A* is not satisfactory for summer cooling work.

In the case of arrangement *B* the air flow in the living zone was directed upwards to the ceiling in a path that did not extend beyond approximately 30 in. from the sidewall. Hence, only a very small portion of the living zone

was subjected to the direct action of the air from the registers. The cold air after reaching the ceiling diffused and descended into the living zone without the creation of any objectionable drafts. In this connection the arrangement proved to be the most satisfactory of the three tested.

The sidewall register located at the 7 ft level, indicated by arrangement *C* in Fig. 11, projected the air outwards into the room and hence did not create any objectionable drafts in the living zone, except near the outside wall. It was observed that the cool air descended parallel to the outside wall and produced an appreciable draft in a small zone near this wall. Therefore, the only limitation to be observed in connection with the use of arrangement *C* is that the outlet register velocity should not exceed a value just sufficient to project the air to the wall opposite to the register. Obviously, the use of diffusing type registers, which spread the air outwards into the room, would tend to minimize this effect. But in the selection of such registers due consideration should be given to the proper relations [10] existing between the air velocity, the air temperature, and the size of the room.

CONCLUSIONS

The following conclusions may be drawn as applying to the Research Residence and the conditions under which the tests were conducted.

(1) Using water at temperatures of 52 F and 46 F, it is possible to maintain an indoor dry-bulb temperature of approximately 80 F with the outdoor temperature as high as 108 F. On days when the maximum outdoor temperature exceeds 90 F the plant will operate continuously over considerable periods of time and it is possible to maintain an indoor relative humidity of 52 per cent, with a resultant effective temperature of approximately 74 deg, which would undoubtedly be acceptable to the majority of users.

(2) During intermittent operation of the cooling plant the indoor relative humidity rises to a maximum during the *off* period. The magnitude of this rise is dependent to a great extent on the moisture content of the outdoor air.

(3) During continuous operation of the cooling plant the indoor relative humidity decreases to a minimum value which is independent of the moisture content of the outdoor air and is related to the mean coil temperature. Possibly by coincidence the numerical value of the relative humidity that was maintained in the house was nearly the same as the numerical value of the mean coil temperature.

(4) In the case of uninsulated frame structures similar to the Research Residence the more complicated method for calculation of the maximum value of the cooling load, based on including the heat lag, may not be justified.

(5) The absence of objectionable drafts in the living zone was found to be a more satisfactory criterion for evaluating the performance of different types of registers used in connection with summer cooling work than was the existence of small temperature gradients from the floor to the ceiling.

ACKNOWLEDGMENTS

The results presented in this paper were obtained in connection with the summer cooling investigation (1936) in the Research Residence [11] at the

[10] Characteristics of Registers and Grilles, J. H. Van Alsburg, A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 245.

[11] The Research Residence in Urbana, Ill., was built, furnished, and completely equipped specifically for research work in warm air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

University of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, Dean of the College of Engineering, is the director, in the Department of Mechanical Engineering of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the head. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is also due to M. K. Fahnestock, Research Assistant Professor, and R. B. Engdahl, Research Assistant, for services rendered in connection with the investigation, and to the various manufacturers who cooperated by furnishing instruments and equipment.

DISCUSSION

PROF. T. F. ROCKWELL (WRITTEN): The author made the statement that sun effect on walls can be neglected without introducing a serious loss in heat gain calculations. In the section devoted to the register tests, it is suggested that arrangement *B* is satisfactory for cooling.

In connection with the first suggestion, there is some tendency to place undue emphasis on heat gain through walls because of sunshine.

As to the second, architects and owners as groups both prefer baseboard registers to those placed in the wall 7 or 8 ft above the floor. Hence, if the authors have any data on the performance of registers under arrangement *B* during the heating season, it would be interesting to have this information discussed.

JOHN EVERETTS, JR. (WRITTEN): I would like to take this opportunity to congratulate the authors for their excellent preparation of this paper. Few of us realize the actual work involved in taking data, segregating it, calculating the innumerable quantities and values required and correlating these values into tangible words and figures which result in the making of a paper which is as valuable to the Society as this one.

The title of the paper is misleading from the standpoint of the actual scientific research activities of the Society, in that it centers, in large part, around the selection of the type of apparatus required to produce the refrigerating effect used in the research work. It has been my contention for a long time that this is a very interesting problem from a commercial angle but hardly a scientific research problem of a technical society.

The portion of this paper which I feel is most valuable, and on which so little work has been done, is the comparison of the calculated cooling load and the actual cooling load due to the variation of the various heat sources and their relation to time lag, shown in Figs. 5a and 5b. As the relation of the actual cooling load to the calculated load is a function of building construction, knowing the design conditions of temperatures and sun effect, a system may readily be designed to economically condition a residence in any locality with the accurate data available and published in THE GUIDE. The fact that an insulated house has a greater time lag than an uninsulated one, and that of a brick construction is greater than frame construction holds true in the same relative proportion whether the house is in Old Greenwich, Conn., or Pasadena, Calif. If we assume the Research Residence in Urbana as a standard (as there is more accurate data available on this than any other) then we may relate houses of different construction on a percentage basis and apply this percentage to the calculated cooling load as based on the methods of calculation given in THE GUIDE. In this way specific standards, regardless of locality, may be set up to give the home owner a job of proper capacity to meet the requirements and will tend to eliminate the unscrupulous practice of installing a system of inadequate capacity purely because the Research Residence of

14,170 cu ft can be cooled with two tons of refrigeration effect. With the cooperation of the various manufacturers who have established model homes with air conditioning in various parts of the country it would not be difficult to establish standards of this type, provided, of course, test data such as are given in this paper are available for proper analysis.

The second item of importance in this paper is the question of air distribution. As the Society has a committee to make a special study of this important question I feel it is more appropriate to hear from them on this subject.

E. V. FINERAN (WRITTEN): In the discussion of Operating Characteristics there appears the following statement, "that is, in both series of tests, the indoor dry-bulb temperature, the indoor relative humidity, and the indoor effective temperature respectively were maintained at approximately the same values, irrespective of the temperature of the water entering the coil." As is shown later on in the paper the relative humidity is affected by the water temperature so that the above statement is misleading. It should be pointed out that the slight difference in water temperature produced an inappreciable difference in the humidity conditions.

Were any data obtained which would establish whether a relation existed between the amount of air infiltration and the amount of outside air introduced through the system?

Was any attempt made to correlate the load to a combination of degree-hours and grain-hours?

Perhaps by trial and error slight changes in the factors for computing the various loads might result in a set of values which would agree more closely with the measured load for all conditions of temperature, sun effect, etc.

Perhaps from the amount of increase in temperature of the building some data might be put in a usable form on the effect of the heat capacity of the building on equipment size.

H. M. BETTS (WRITTEN): The authors of this paper should be commended for their very excellent presentation of their findings in their continued research in the field of residential cooling. Coming as it does at a time when the public is showing a decided interest in this subject, the data contained in this paper should prove of immense value to those interested in such equipment.

With regard to cooling load calculations a comparison of the curves in Figs. 5a and 5b indicates that for average residential construction the time at which the maximum cooling load occurs is relatively unimportant. Thus, assuming abnormal occupancy loads are not to be considered, the consideration of time lag could be dispensed with in cooling load calculations for this type of building.

One of the most important conclusions arrived at as a result of this investigation is in regard to location of registers. Referring to Figs. 10 and 11 it is apparent that the conventional baseboard register when used in connection with a cooling system gives results that are anything but desirable. It must be remembered that by far the largest percentage of mechanical warm air heating plant installations of the past few years have baseboard registers which are little more than bordered openings having a few vertical bars sufficient to restrain the nocturnal wanderings of the family cat. Since it is installations of this type which are the potential cooling jobs of tomorrow, it is apparent that unless registers are changed to the type indicated in curve "B" of Fig. 10 or are altered to give better diffusion upwards, nothing but a dissatisfied customer can be the result. The *National Warm Air Heating and Air Conditioning Association* should make every effort to place copies of this paper in the hands of their members with instructions to read and digest its contents thoroughly for their own good and for the good of the industry.

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PROF. A. P. KRATZ: Reference has been made to the formula used by F. H. Faust and others for the calculation of the heat gain by walls exposed to the sun. It should be understood that this method was essentially the same as the method used by us. Both methods are based on the same fundamental equations and upon the same assumptions. Our formula factors to the form:

$$H = U(t_o - t_i) + \frac{UeI}{f_o}$$

in which H = heat gain, Btu per square foot per hour.
U = coefficient of heat transmission, Btu per square foot per hour per degree temperature difference from air to air.
f_o = outside surface coefficient, Btu per square foot per hour per degree temperature difference from surface to air.
t_o = outdoor air temperature, F.
t_i = indoor air temperature, F.
I = sun's intensity on the proper vertical surface, Btu per square foot per hour.
e = emissivity coefficient for the sun's radiation.

This is in the same form as that given by Mr. Faust and the factor $\frac{U}{f_o}$ is the same as his factor F which he evaluated by means of a curve.

No. 1063

COOLING LOAD ANALYSIS OF A BANK BUILDING

By JOHN JAMES* (*MEMBER*), NEW YORK, N. Y.

HOW do design engineers appraise the cooling load factors of solar radiation, uncontrolled infiltration and controlled ventilation? What is customary practice in determining cooling load requirements for the selection of equipment? When are safety factors used and how much? Why are bidders' proposals so far apart for air conditioning a given enclosure? How will future research investigations tend to eliminate many of the unknown variables? Answers to these important questions are discussed in this paper, based on a survey conducted in 27 of the United States and several provinces of Canada and the results are particularly significant.

Load factors for heating are fairly well established, but scientific data are still lacking for accurately estimating the several components of the cooling load, which accounts in part for the lack of uniformity in cost estimates and system designs. Practically every engineer and manufacturing organization has devised forms for calculating a cooling load, and it is interesting to note that many of them are based on Society research data. Some forms of computation are very comprehensive taking into consideration all of the various components of the load as they are affected by the particular building design. Other forms are based on empirical values which are assumed to be typical or average figures that can be used on all types of building construction with sufficient accuracy.

SCOPE OF SURVEY

A method of cooling load computation is outlined in Chapter 8 of THE A.S.H.V.E. GUIDE, 1937, which is based on numerous investigations conducted at the Research Laboratory and cooperative institutions. A typical one story banking space problem shown in Fig. 1 was forwarded to members of the Society with a request that a sample cooling load calculation be prepared in accordance with their usual practice. It was thought that a survey of this kind would indicate whether THE GUIDE method was actually being used in the field and, if not, what other procedures were in general practice.

In response to an invitation extended to engineers in every state and prov-

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ince of Canada, 47 calculations were returned representing most of the principal territories requiring summer air conditioning. These replies were submitted from technical men of the following professional status: 21 consulting

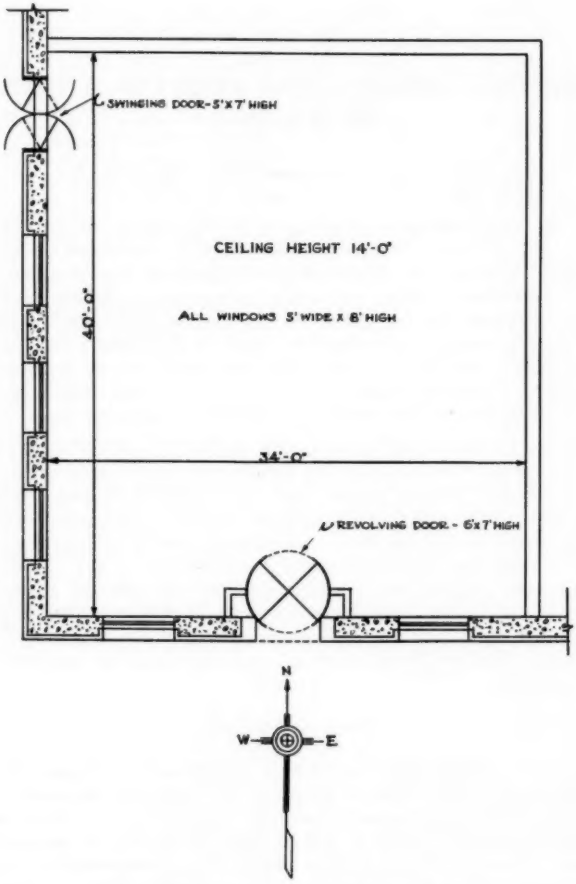

FIG. 1. PLAN DIAGRAM OF BANKING SPACE

engineers, 11 manufacturer's engineering representatives, 8 utility engineers, 4 contractors and 3 professors.

It was the intent of the survey to limit the study primarily to factors affecting the solar radiation transmitted, uncontrolled infiltration air, and controlled

ventilation air. Hence, it was found desirable to reduce the calculations to a minimum of variables with general design conditions as follows:

Building Construction:
Outside Wall, $U = 0.37$ for 4 in. red brick veneer, backed with 16 in. concrete and ½ in. plaster.
North and East Inside Walls—no heat gain.
Flooring—no heat gain.
Roof, $U = 0.26$, for 2 in. concrete roof deck with black finish roofing material, ½ in. rigid insulation and suspended plaster ceiling.
Windows, $U = 1.13$ for single glazed double-hung metal non-weatherstripped.
People load, 26 individuals.
Light load, 4,500 watts.
Assume no shading effects from adjacent structures in calculating sun effect.

In order to unify the tabulation of the results, it was requested that the calculation sheet show individual determinations of the sensible, latent and total cooling loads. So that values in THE GUIDE could be compared, it was suggested that the design inside and outside dry- and wet-bulb temperatures be given. Other significant items requested in the statement of conditions were a complete explanation of the assumptions used in determining the solar radiation effects through windows, vertical walls and roof. Also, the stated amount of outside air introduced for ventilation was to be given with values for the air infiltration through the windows, swinging doors and revolving doors.

SAMPLE LOAD CALCULATION

Using the stated conditions for the problem and assuming certain design conditions, a sample calculation of the cooling load is indicated in Table 1 based on the method outlined in THE GUIDE 1937. The computation is divided into the various components of both the sensible and latent heat gains. Transmission and sun effect are determined from the degrees which are recommended to be added to the outside dry-bulb temperature for different types of surfaces. THE GUIDE indicates methods for the determination of infiltration based on the amount of air per lineal foot of window crack and also on the number of air changes taking place depending on the kind of room or building. Checking the total amounts of air entering the enclosure by these two methods shows that the results in each case are less than the specified amount recommended in Chapter 3 of THE GUIDE for the quantity of outside air introduced for ventilation to take care of the human occupancy requirements. Therefore, it is assumed in this case that the ventilation load is based on the controlled outside air, thus resulting in a positive pressure within the enclosure and causing exfiltration through the window and door openings.

DESIGN OUTSIDE AND INSIDE CONDITIONS

A primary consideration in a cooling load determination is the selection of the correct outside and inside dry- and wet-bulb temperatures. Frequently, engineers in the same locality have divergent views in respect to the proper temperatures to assume. Values given in THE GUIDE for heating load calculations are merely a compilation of climatic conditions obtained from government Weather Bureau records and even though the lowest temperatures are

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stated, engineers rarely base their design on these values. Such temperatures are usually of short duration and are rarely repeated in successive years. When designing a heating system it is customary practice to assume an outside temperature of 15 F above the lowest recorded temperature.

Likewise, in the design of a cooling system it is important to exclude these maximum outside temperatures which occur for short periods of duration.

TABLE 1—SAMPLE COOLING LOAD CALCULATION BASED ON "THE GUIDE" METHOD

DESIGN CONDITIONS			
Outside	100 F dry-bulb	75 F wet-bulb	87 grains per pound dry air
Inside	80 F dry-bulb	67 F wet-bulb	77 grains per pound dry air

SENSIBLE HEAT GAIN		**BTU PER HOUR**
Transmission and Sun Effect		
South Wall	$= 354 \times 0.37 \times [(100 + 10) - 80]$	$= 3,930$
West Wall	$= 405 \times 0.37 \times [(100 + 20) - 80]$	$= 6,000$
North Wall, no heat gain		
East Wall, no heat gain		
South Glass	$= 122 \times 90$ (2 p. m.) $\times 0.97$	$= 10,680$
West Glass	$= 155 \times 140$ (2 p. m.) $\times 0.97$	$= 21,000$
Roof	$= 1,360 \times 0.26 \times [(100 + 45) - 80]$	$= 23,000$
People Load	$= 26 \times 220$	$= 5,720$
Lighting Load	$= 4,500 \times 3.413$	$= 15,300$
Ventilation, assume 15 cfm per person outside air introduced with exfiltration due to positive fan pressure of system	$V_s = (15 \times 26) \times 60 \times 0.24 \times 0.0735 \times (100 - 80)$	$= 8,250$
	Total Sensible Heat Gain =	93,880
LATENT HEAT GAIN		
People Load	$= 26 \times 180$	$= 4,680$
Ventilation V_l	$= (15 \times 26) \times 60 \times 0.0735 \times \frac{87-77}{7,000} \times 1,060$	$= 2,600$
	Total Latent Heat Gain =	7,280
TOTAL COOLING LOAD		101,160
Tons of Refrigeration	$101,160 \div 12,000 = 8.42$	

Values given in Table 1, page 158, of THE GUIDE 1937 are design temperatures which are based on a study of hourly temperatures in New York City from which factors were derived and applied to the average maximum dry- and wet-bulb temperatures for other cities. In some cases revisions have had to be made in the original table based upon the recommendations of committees appointed to study weather conditions by the local Society Chapters.

It should be recognized that the type of building occupancy governs in many respects the selection of proper outside design temperatures. The values given in THE GUIDE are for an installation operating approximately 10 light hours a day or for a cooling season of 1,200 hours in June, July, August and September for an average year. If the system is being designed for an industrial plant operating 24 hours a day or for a theater operating from 11:00 a.m. to 11:00

TABLE 2—COMPARATIVE DESIGN OUTSIDE AND INSIDE CONDITIONS FOR COOLING LOAD DETERMINATIONS

No.	State	City	The Guide 1937 Table 1 Chap. 8 Outside Temp. Deg F		Analysis Outside Temp. Deg F		Analysis Inside Temp. Deg F		Eff. Temp.	R. H. %
			D. B.	W. B.	D. B.	W. B.	D. B.	W. B.		
1	Ala.	Birmingham	93	77	95	78	80	67.0	74.0	50
2	Calif.	Los Angeles	88	70	95	74	75	62.5	70.0	50
3		Los Angeles	88	70	90	70	77	64.5	71.7	50
4		Fresno	.....	.....	100	69	80	65.5	73.4	45
5		Oakland (San Francisco)	85	68	85	67	75	63.0	70.1	50
6	D. C.	Washington	95	78	95	78	80	65.0	73.2	44
7	Fla.	Miami Beach (Tampa)	94	79	91	79	79	67.0	73.6	53
8	Ill.	Chicago	95	75	95	75	80	67.0	74.0	50
9		Chicago	95	75	95	75	80	67.0	74.0	50
10		Chicago	95	75	95	75	80	67.0	74.0	50
11		Urbana (Peoria)	91	75	91	75	80	67.0	74.0	50
12	Ind.	Indianapolis	90	73	95	77	78	65.0	72.3	50
13	Iowa	Des Moines	92	74	92	74	78	64.5	72.1	49
14	Ky.	Lexington (Louisville)	94	75	94	75	82	67.0	75.0	45
15	La.	New Orleans	94	79	95	79	80	67.0	74.0	50
16		New Orleans	94	79	95	79	80	67.0	74.0	50
17	Md.	Baltimore	93	76	95	78	80	67.0	74.0	50
18		Baltimore	93	76	95	78	80	67.0	74.0	50
19	Mass.	Boston	88	73	90	75	78	66.0	72.7	54
20	Mich.	Detroit	93	73	95	75	80	67.0	74.0	50
21		Grand Rapids	.....	.....	95	75	80	67.0	74.0	50
22		Kalamazoo	.....	.....	95	78	80	67.0	74.0	50
23	Mo.	Kansas City	92	75	100	75	80	67.0	74.0	50
24		Kansas City	92	75	100	78	80	67.0	74.0	50
25	Mont.	Billings (Helena)	87	63	100	71	82	67.0	75.0	45
26	Nebr.	Omaha (Lincoln)	93	74	95	78	80	67.0	74.0	50
27	N. J.	Newark (Trenton)	95	76	95	79	80	65.0	73.2	44
28	N. Y.	Buffalo	83	72	95	75	80	67.0	74.0	50
29		New York	95	75	95	75	80	67.0	74.0	50
30	N. D.	Bismark (Grand Forks)	88	69	86	68	77	64.5	71.7	50
31	Ohio	Cincinnati	95	78	95	78	80	67.0	74.0	50
32		Cincinnati	95	78	95	78	80	65.0	73.2	44
33		Cleveland	95	75	95	75	80	67.0	74.0	50
34		Columbus	.....	.....	95	75	80	66.0	73.6	47
35		Columbus	.....	.....	95	77	80	65.0	73.2	44
36		Dayton	.....	.....	95	78	80	67.0	74.0	50
37	Okla.	Oklahoma City	96	76	96	76	81	65.9	74.0	44
38	Oreg.	Portland	83	65	85	65	75	65.0	70.8	59
39	Pa.	Pittsburgh	91	73	90	73	78	64.0	72.0	46
40		Pittsburgh	91	73	95	75	80	67.0	74.0	50
41	Tenn.	Memphis	93	77	95	78	80	67.0	74.0	50
42	Texas	Ft. Worth (Dallas)	99	76	100	78	83	67.0	75.5	43
43	Vt.	Burlington	85	71	90	73	80	68.0	74.5	55
44	Wash.	Seattle	83	61	84	63	70	60.0	66.4	56
45	Wis.	Milwaukee	93	74	95	75	80	67.0	74.0	50
46	Canada	Winnipeg	.....	.....	90	70	76	66.0	71.8	59
47	Canada	Toronto	.....	.....	90	72	75	66.0	71.1	61

p.m., it is obvious that a study of the hourly temperatures would show different results because of the cooler night air conditions.

Outside Design Temperatures

Based on the results obtained from this survey, values are given in Table 2 showing the design outside and inside temperature conditions for various localities. The first two columns of the tabulation show figures corresponding to the values given in THE GUIDE and the latter columns refer to the results re-

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ported in an analysis of the survey. In many instances the temperatures obtained from the survey check closely with THE GUIDE values and the extreme variations occur only in the following cities: Los Angeles, Indianapolis, Baltimore, Boston, Detroit, Kansas City, Billings, Omaha, Newark, Buffalo, Pittsburgh, Ft. Worth, Burlington and Seattle.

A Joint Committee [1] of the A.S.H.V.E. and *American Society of Refrigerating Engineers* recently considered a re-study of design outside temperatures for heating and cooling load estimates and as a result a Society Research Technical Advisory Committee was appointed to develop a formula for appraising the proper design conditions from Weather Bureau records. Upon completion of a satisfactory formula or standard to use for this investigation, it is

TABLE 3—RECOMMENDED INDOOR AIR CONDITIONS IN SUMMER CORRESPONDING TO OUTDOOR TEMPERATURES

OUTSIDE DESIGN DRY-BULB TEMP. DEG F	INSIDE DESIGN EFFECTIVE TEMPERATURES	
	THE GUIDE[a] 1937 Table 2, Chap. 3	Table in Proposed Code[b] of Min. Requirements for Comfort Air Cond.
105	.....	75.5
100	.....	75.0
95	73.0	74.0
90	72.0	73.0
85	71.0	72.0
80	70.0	71.0
75	69.0	.....
70	68.0	.....

[a] Applicable to exposures less than 3 hours.
[b] For occupancies in excess of 40 min and for sedentary occupations between relative humidity limits of 30 and 70 per cent.

expected that the Research Laboratory will arrange for a comprehensive survey of all Weather Bureau records in the principal cities of this country and Canada.

Inside Design Temperatures

The problem of selecting the proper indoor air conditions for a given enclosure is one which is open to considerable discussion and debate. Recent research seems to indicate that effective temperature is important only from a standpoint of comfort and that the dry- and wet-bulb temperatures may vary over a considerable range if the proper effective temperature is maintained within limits of the summer comfort zone. Several references are given in Chapter 3 of THE GUIDE outlining the selection of the proper effective temperature to use in the design of a system for buildings of various occupancies. For several years data have been included in THE GUIDE, shown in the second column of Table 3 of this paper, on recommended indoor air conditions in summer corresponding to various outdoor temperatures. THE GUIDE values

[1] A.S.H.V.E.-*A.S.R.E.* Committee on National Standards for Air Conditioning Applications.

have been subject to much discussion but up to the present time no satisfactory substitute has been offered. Many uncontrollable variables enter into this problem and it is expected that work now being conducted at the Research Laboratory and other interested institutions may throw considerable light on this complex problem.

In the last column of Table 3 are values suggested in a proposed Code now available for discussion as submitted by the Joint Committee.[2] It states on page 72 of THE GUIDE 1937, for banks and stores where the period of occupancy is short, that the contrast between outdoor and indoor air conditions becomes the deciding factor in regard to the maintenance of proper tempera-

TABLE 4—DESIGN INSIDE TEMPERATURES CORRESPONDING TO OUTSIDE TEMPERATURES

OUTSIDE DESIGN DRY-BULB TEMP. DEG F	NUMBER INSIDE DESIGN EFFECTIVE TEMPERATURES CORRESPONDING TO OUTSIDE DESIGN DRY-BULB TEMPERATURES															
	75.5	75.0	74.5	74.0	73.6	73.4	73.2	72.7	72.3	72.1	72.0	71.7	71.1	70.8	70.1	70.0	66.4
100	1	1		2		1											
96				1													
95				20	1		5		1							1	
94		1															
92										1							
91				1	1												
90			1					1			1	1	1				
86												1					
85														1	1		
84																	1

ture and humidity. General experience favors a schedule of design for banks about 1 deg effective temperature higher than that shown in column 2 of Table 3. It will be noted that if this recommendation is followed the values proposed for the Code would check quite closely with the figures now given in THE GUIDE.

Table 4 is arranged so as to give the number of inside design effective temperatures for various cities corresponding to the outside design dry-bulb temperatures as taken from Table 2. With only outside design dry-bulb temperatures used as a basis of comparison, it will be noted that the survey indicated a close check with values in Table 3 for conditions corresponding to 95 F outside and 74 deg inside effective temperature.

CLIMATIC CONDITIONS IN OTHER LOCALITIES

Incomplete data were supplied by a few engineers to comprise a complete analysis for the banking problem but the information forwarded regarding

[2] Loc. Cit. See Note 1.

climatic conditions is of particular significance in connection with this discussion.

Philippine Islands

Complete temperature data seem to be lacking in Manila, but general observations show wet-bulb temperatures of from 77 to 79 F with maximum dry-bulb temperatures varying between 80 and 97 F. Outside design conditions for this locality are generally assumed to be based on 93 F dry-bulb and 79 F wet-bulb for cooling load determinations. Desirable indoor conditions are selected on a basis determined by taking two-thirds of the difference between the outdoor temperature and 75 F dry-bulb with relative humidity limits of 40 to 60 per cent. Recommended inside temperatures corresponding to outside conditions are shown in Table 5.

Denver, Colorado

Engineers familiar with sea level atmospheric conditions rarely appreciate the design differences which exist at high altitudes when applied to the selection of air conditioning systems. Information obtained from Denver emphasizes that the mountain states of that region have a wet-bulb temperature not over 60 F practically 95 per cent of the time. Summer temperature conditions of 95 F dry-bulb and 60 F wet-bulb are apparently not uncommon in that locality with the Weather Bureau frequently reporting 5 to 10 per cent relative humidities with 95 to 100 F dry-bulb late in the afternoon during days in July and August.

With barometric pressures in Denver of 24.65 in. of mercury it is highly important to consider the differences which exist in vapor pressure, atmospheric moisture content, specific volume and heat content of the air. From charts prepared for this altitude, it is interesting to note that at an air temperature of 84 F saturation one pound of air holds approximately 40 grains more moisture than at sea level.

Air conditioning loads calculated for Denver should also consider the extraordinary amount of solar radiation which is present at high altitudes as the added amounts in heat gain from this source are of prime importance.

SOLAR RADIATION FACTORS

After the selection of the proper outdoor and indoor temperature conditions, one of the next steps in an analysis of a cooling load is to determine the transmission of heat through the walls, roof and glass. Transmission coefficients are available for almost all of the common building materials and insulators and consequently little variation in final results is obtained from this phase of a cooling load determination.

The solar radiation through walls, roof and glass as used in the various computations of this survey is given in Table 6. It is apparent that many engineers are using the method outlined in THE GUIDE which provides for the addition of a certain number of degrees to the outside dry-bulb temperature for different surfaces. Table 2, appearing on page 160 of THE GUIDE 1937

and based on a number of A.S.H.V.E. papers,[3] was prepared several years ago to provide a simplified method of allowing for sun effect. It is recognized that the calculations for the transmission of heat from solar radiation through building walls are quite complicated due to the periodical character of heat flow and the time lag or heat capacity of construction. Consequently, the present GUIDE method was introduced to be of practical value to the engineer who was unfamiliar with the many variations in solar heat transmission.

Although values of temperature increase for glass are shown in column 23, it should be noted that this method is not suggested in THE GUIDE. Appearing in column 14 are values of solar intensity obtained from curves similar to those on page 161 of THE GUIDE 1937, which should be used with the wall and roof values of columns 22 and 24 for a computation of this form. Percentage

TABLE 5—OUTSIDE AND INSIDE TEMPERATURES FOR MANILA, PHILIPPINE ISLANDS

OUTSIDE TEMP. DEG F		INSIDE TEMP. DEG F	
Dry-Bulb	Wet-Bulb	Dry-Bulb	Wet-Bulb
96	76.0 to 79.0	82	66.0 to 67.0
93	76.0 to 79.0	81	66.0 to 67.0
90	76.0 to 79.0	80	66.0 to 67.0
87	75.5 to 78.5	79	65.5 to 66.5
84	75.0 to 78.0	78	65.5 to 66.5
81	74.0 to 77.0	77	65.5 to 66.5
78	73.0 to 76.0	76	65.0 to 66.0

values are indicated in column 15 depending upon whether the designer assumed a bare or shaded window.

Occasionally engineers use in addition to the solar intensity and shading factor previously mentioned for glass, the heat transmission coefficient and the temperature difference, all as applied to the surface area. This gives an additional heat gain which is unwarranted as the shading factor is a test value which takes into consideration the glass transmission.

In an effort to reduce the selling cost of air conditioning equipment, many manufacturers as well as engineers designing systems have devised empirical methods of computation which incorporate in a single value, factors for heat transmission, and solar radiation. Included in columns 19, 20 and 21 of Table 6 are values of this kind and although admittedly simpler to use, it is readily recognized that they are not based on a scientific approach to the problem. In some cases the values given in these columns are based on the building construction of the job and vary depending upon the conditions stated. When the values are obtained in this manner the results are satisfactory. Considerable engineering judgment is required in using figures of this kind and obvi-

[3] Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932). Absorption of Solar Radiation in Relation to the Temperature, Color, Angle, and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930). Coefficients of Heat Transfer as Measured under Natural Weather Conditions, by F. C. Houghten and C. G. F. Zobel (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).

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TABLE 6—COMPOSITE TABULATION OF BANK

No.	State	City	Cooling Load Tons of Refrigeration			Safety Factors in Per Cent			Methods of Comput-								
									Solar Intensities and Related Factors Used with Surface Areas[a]								
			Tot'l	Sensible	Latent	Tot'l	Sensible	Latent	Wall Direction	Wall			Glass		Roof		
										I	*e*	*F*	*I*	*f*	*I*	*e*	*F*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
1	Ala.	Birmingham	9.10	8.07	1.03				West								
2	Calif.	Los Angeles	12.70	8.95	3.75				West								
3		Los Angeles	7.66	6.95	0.71				West								
4		Fresno	8.17	8.17	0.00		4		West								
5		Oakland	7.12	6.44	0.68		4		West								
6	D. C.	Washington	11.50	8.43	3.07				South West	45 185	0.7 0.7	.085 .085	20 165		200	0.9	.060
7	Fla.	Miami Beach	6.66	4.66	2.00	10	10	10	South West								
8	Ill.	Chicago	7.70	5.71	1.99		10		South West								
9		Chicago	7.77	6.73	1.04		10	5	West	50	0.7	.09	130		100	0.9	.060
10		Chicago	10.75	9.33	1.42				South West								
11		Urbana	9.35	8.25	1.10				South West	90 135	0.7 0.7		90 135	97 97	255	0.9	
12	Ind.	Indianapolis	9.44	8.04	1.40				South West	45 185	0.7 0.7	.087 .087	20 165		205	0.9	.062
13	Iowa	Des Moines	8.89	7.81	1.08				West								
14	Ky.	Lexington	10.22	7.38	2.84				West				150				
15	La.	New Orleans	6.47	4.89	1.58		4		West	185	0.7	.061	165				
16		New Orleans	8.33	6.61	1.72		10		West				80				
17	Md.	Baltimore	9.06	7.74	1.32		10	10	South West				100 100				
18		Baltimore	8.58	7.13	1.45				South West				60 155				
19	Mass.	Boston	8.01	7.18	0.83		15	10	South West				41 117				
20	Mich.	Detroit	5.91	5.21	0.70		5	5	West								
21		Grand Rapids	7.75	6.50	1.25		4		West				150				
22		Kalamazoo	8.30	7.34	0.96	10	10		West								
23	Mo.	Kansas City	8.49	7.82	0.60				South West				90 140	97 97			
24		Kansas City	9.24	8.30	0.94				West								
25	Mont.	Billings	7.57	6.94	0.63				South West								

[a] Symbol reference notations: *I* = solar intensity normal to surface. Btu per square foot per hour; *f* = percentage solar radiation transmitted through bare and shaded windows.

BUILDING COOLING LOAD DETERMINATIONS

ING SOLAR HEAT GAIN						HEAT GAIN ALLOWANCES FOR INFILTRATION AND VENTILATION AIR ENTERING ENCLOSURE			
Empirical Values Used with Surface Areas			Degrees Added to Outside Dry-Bulb Temp.			Uncontrolled Infiltration		Controlled Outside Air for Ventilation	Column Nos. Used in Computation
Wall	Glass	Roof	Wall	Glass	Roof	Window Air Quantity or Room Air Changes per Hour	Swinging (S) and Revolving (R) Door Air Quantity		
19	20	21	22	23	24	25	26	27	28
12.8	190	16				40–60 cfh/sq ft	S-60–80 cfh/sq ft R-40–60 cfh/sq ft	20 cfm/person	27
5.5	100	5.2				1.5 air change		⅓ total air	25+27
3.7	180	7.8				2 air change			25
3.7	180	7.8				2 air change			25
3.7	180	7.8				2 air change			25
							320 cfm, based on 200 passages/hour	15% total	26+27
			13 28	5 5	38		800 cfm total	20 cfm/person	27
			5 −5	0 43.4	35	20 cfm/sq ft	20 cfm/sq ft	25% total	25+26+27
						15 cfh/sq ft	S-30 cfh/sq ft R-30 cfm x 3 per x 60 = cfh	15/cfm person	25+26+27
6.0 0	139 183	60				50 cfh/ft crack using wallhavingmostcrack	200 cfh/ft crack for 1 set of doors	15 cfm/person 10% total	27
						45 cfh/ft crack using wallhavingmostcrack	S-16 cfm/person R-6.5 cfm/person	15 cfm/person	25+26+27
						0.78 cfm/ft crack	S-1.56 cfm/ft crack S-8 cfm/person R-6.5 cfm/person	10 cfm/person	25+26
	136	27.9				1 air change			25
			20		20	6 air change			25
					5		S-8 cfm/person R-6.5 cfm/person	20 cfm/person	27
					45	1.5 air change		10 cfm/person	25+27
			−5 −5		35	0.5 air change		1 air change	25+27
			10 10		45	0.5 air change		1 air change	25+27
			0 5		43		S-8 cfm/person R-6.5 cfm/person	10 cfm/person	27
8.2	139					0.78 cfm/ft crack 0.62 cfm/sq ft	1 air change	10 cfm/person	26
			20		20	Infiltration by GUIDE Method	Not less 1 air change	10 cfm/person	25+27
5.5	60	9.1						20% total	27
			10 20		45			15 cfm/person	27
3.0	100	12.0						15 cfm/person	27
0 0	35 165	13.05						20 cfm/person	27

e = coefficient of absorption of solar radiation; F = percentage solar radiation transmitted through wall;

TABLE 6 (*Continued*)—COMPOSITE TABULATION OF BANK

No.	State	City	Cooling Load Tons of Refrigeration			Safety Factors in Per Cent			Methods of Comput-									
									Solar Intensities and Related Factors Used with Surface Areas[a]									
			Tot'l	Sensible	Latent	Tot'l	Sensible	Latent	Wall Direction	Wall			Glass		Roof			
										I	*e*	*F*	*I*	*f*	*I*	*e*	*F*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
26	Nebr.	Omaha	9.48	7.54	1.94				South	65	0.7	.087	122	68			
									West	180	0.7	.087	155	68	195	0.9	.060
27	N. J.	Newark	8.46	5.82	2.64	10			South								
28	N. Y.	Buffalo	7.39	5.96	1.43				South				90	68			
									West				140	68			
29		New York	7.15	5.37	1.78				West								
30	N. D.	Grand Forks	8.33	7.27	1.06				South				80				
									West				210				
31	Ohio	Cincinnati	7.04	5.13	1.91				South								
									West				120				
32		Cincinnati	10.72	6.64	4.08		5		West				79	58			
33		Cleveland	8.00	6.63	1.37	10	5		South				75				
									West				100				
34		Columbus	6.95	5.90	1.05				West				100				
35		Columbus	8.55	7.50	1.05				South				62	97			
									West				185	97			
36		Dayton	5.90	5.06	0.84				South								
									West								
37	Okla.	Oklahoma City	9.00	7.53	1.47				South				55	97			
									West				207	97			
38	Ore.	Portland	7.19	6.78	0.41		10	5	South	90	0.7	.085	90	97			
									West	140	0.7	.085	140	97	250	0.9	.060
39	Pa.	Pittsburgh	8.80	7.58	1.22	10			West								
40		Pittsburgh	10.45	8.36	2.09				South				100				
									West				115				
41	Tenn.	Memphis	5.23	4.43	0.85				South								
									West								
42	Texas	Ft. Worth	8.04	6.33	1.71				West				80				
43	Vt.	Burlington	6.90	6.21	0.69		5	5	West				180				
44	Wash.	Seattle	8.03	7.03	1.00	10	10		South				80	60			
									West				138	60			
45	Wis.	Milwaukee	7.99	6.98	1.01	10	10		South				17				
									West				125				
46	Canada	Winnipeg	7.42	7.04	0.38				South				73				
									West				84				
47		Toronto	10.36	9.79	0.57				South				161				
									West				183				

[a] Symbol reference notations: I = solar intensity normal to surface. Btu per square foot per hour; f = percentage solar radiation transmitted through bare and shaded windows.

BUILDING COOLING LOAD DETERMINATIONS

ING SOLAR HEAT GAIN						HEAT GAIN ALLOWANCES FOR INFILTRATION AND VENTILATION AIR ENTERING ENCLOSURE			
Empirical Values Used with Surface Areas			Degrees Added to Outside Dry-Bulb Temp.			Uncontrolled Infiltration		Controlled Outside Air for Ventilation	Column Nos. Used in Computation
Wall	Glass	Roof	Wall	Glass	Roof	Window Air Quantity or Room Air Changes per Hour	Swinging (S) and Revolving (R) Door Air Quantity		
19	20	21	22	23	24	25	26	27	28
								30 cfm/person	27
	40	10						25% total	27
			10 20		45	75% of 1.5 air change		15 cfm/person	25+27
			25	25	40			25% total	27
			10 20		45	1 air change		1.5 air change	25+27
			10		45	0.5 air change		1.5 air change	25+27
			5		45	45 cfh/ft crack	S-17,000 cfh R-3,600 cfh	1 air change	25+26+27
			15		25	1 air change		10 cfm/person	25+27
.407	1.234	.286	5		30	17 cfh/sq ft or 1.5 air change	90 cfh/sq ft	500 cfm	27
			10 20		60	40 cfh/ft crack	R-200 entrances/ hour and 40 cu ft/ entrance	10 cfm/person	27
9.0 12.0	45 67	9.0						10 cfm/person	27
			11 21		25	2 air change			25
								15 cfm/person	27
3.0	130	12.0				45 cfh/ft crack	S-150 cfh/ft crack R-200 cfh/ft crack	400 cfm	25+26+27
			10 20		41			25% total	27
3.0 3.0	30 30	3.0						10 cfm/person	27
			20		45	1.5 air change			25
			10		30	1.25 air change			25
			6 1		31		S-8,400 cfh R-50 pers/hour/ door vol.	10 cfm/person not less 20% total	26+27
			0 0		55	45 cfh/ft crack	R-total 165 cfm	10 cfm/person	25+26+27
			10 0		57			10 cfm/person	27
			10 0		57	1 air change	S-135 cu ft per pass. for 200 pass./hour R-35 cu ft per pass. for 200 pass./hour	15 cfm/person	26+27

ϵ = coefficient of absorption of solar radiation; F = percentage solar radiation transmitted through wall;

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ously they should only be used in large applications by an engineer with considerable experience.

In 1935 a paper [4] was presented before the Society which outlined a rational method of cooling load determination based on the use of solar radiation factors with various absorption coefficients when applied to walls facing several directions at different latitudes and at different times of the day. It was evident that six of the calculations submitted were based on this method of computation but with variations in some instances. The values given in columns 11 to 18 inclusive of Table 6, together with the footnotes, show in detail the various items considered.

In the cooling load calculation No. 11 for Urbana, Ill., a composite formula [5] for walls exposed to sun was used which incorporated the sun radiation effects in a single equation together with other sensible heat transmission gain factors as follows:

$$H = \frac{AU}{f_o}\left[eI + f_o\ (t_o - t_i) \right]$$

where

H = heat transmitted, Btu per hour.
A = wall area, square feet.
U = over-all coefficient of heat transmission, Btu per square foot per hour per degree Fahrenheit temperature difference from air to air.
f_o = outside surface conductance, Btu per square foot per hour per degree Fahrenheit temperature difference between the surface and air.
e = coefficient of absorption of solar radiation.
I = solar intensity normal to surface, Btu per square foot per hour.
t_o = outdoor air temperature, degrees Fahrenheit.
t_i = indoor air temperature, degrees Fahrenheit.

This equation also takes into consideration the variation in outside surface conductances which the other methods previously described neglected.

Data given in Table 6 show the complete lack of uniformity in the selection of wall surfaces for considering the solar radiation effects. When empirical values were applied it was quite common to select that wall and glass surface having the greatest area irrespective of the compass direction. Other procedures used different solar intensities for the time of day involved as applied to all sun exposed surfaces.

Time Lag and Heat Storage of Walls

Differences in solar intensity values selected for use in the computations were due in some cases to the assumptions made for determining the time of day when the maximum design load occurred.

One customary method of calculation was based on the premise that the changes in load occur coincident with the varying outdoor temperature and sun effect for a particular time of day. Since the walls of a structure offer resistance to heat flow it is obvious that the prevailing outdoor temperature and solar heat are not transmitted instantaneously to the inner surface which

[4] Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads by F. H. Faust, L. Levine and F. O. Urban (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935).
[5] Investigation of Summer Cooling in the Warm-Air Heating Research Residence, by A. P. Kratz, M. K. Fahnestock and S. Konzo. (University of Illinois Engineering Experiment Station Bulletin No. 189).

introduces large errors when this method is applied to walls having high heat storage capacity.

A few computations took into account time lag assuming sun effects and temperature differences existing at some previous time to determine the maximum load. In one instance, the sun load for the roof was completely discounted as the calculator assumed that this peak load did not occur simultaneously with the peak load on the west wall.

UNCONTROLLED INFILTRATION

In general the values stated in Table 6 for determining the amount of uncontrolled air entering the enclosure by infiltration were based on the data given in Chapter 6 of THE GUIDE 1937. In some instances, the calculators used certain established air quantities with surface areas rather than on a basis of the length of crack.

The air change method seemed to be applied to many of the computations varying in amount from half to two per hour and higher according to the judgment of the engineer.

Variations in the amount of infiltration air for the swinging and revolving doors was surprisingly large as is illustrated in column 26 of Table 6. Several values were based on the number of cubic feet of air per passage as directly related to the customer traffic. Others were based on an air quantity per surface area and in a few cases the recommendations were related to the length of door crack. It is apparent that engineers are not in full agreement regarding door infiltration values and with few exceptions little scientific data are available on this subject. A few references in the survey were made to a paper [6] which reported the results of numerous observations conducted on actual installations in the field.

CONTROLLED VENTILATION AIR

The amounts of outside air introduced for ventilation purposes shown in column 27 of Table 6 ranged principally from 10 to 15 cfm per person with one recommendation approaching a high value of 30 cfm per person. The tabulation indicates that in a few cases the air change method was adopted for establishing the ventilation air requirements.

It was frequently mentioned in the computations that the uncontrolled infiltration requirements were determined on a basis of THE GUIDE method for windows and doors along with the air change method, and then compared with the calculated amount of ventilation air. In almost every case the greatest of the three values was used for determining the total cooling load. Column 28 indicates the various air quantities used in the computation and in a few instances it will be observed that all three items were considered.

SAFETY FACTORS

After the total sensible and latent heat quantities were determined several engineers indicated the addition of certain safety factors to take care of the

[6] The Infiltration Problem of Multiple Entrances by A. M. Simpson and K. B. Atkinson (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June 1936).

presence of fan motors within the enclosure, duct gains and related items. Heat gain determinations Nos. 11, 15, 16, 33 and 47 allowed directly for fan motors in the amounts of 1 to 3 hp.

Although not stated specifically, it is probable that some of the safety factors may have been added to take care of starting up loads for the cooling equipment. Standards [7] of one municipality recommended the addition of percentages to the sensible heat load according to the following schedule:

DESIGN SENSIBLE HEAT COOLING LOADS TONS OF REFRIGERATION	PERCENTAGE TO BE ADDED	DESIGN SENSIBLE HEAT COOLING LOADS TONS OF REFRIGERATION	PERCENTAGE TO BE ADDED
0–1	25	10–14	12
2–4	22	14–16	10
4–6	18	16–20	8
6–8	16	20–25	6
8–10	15	25 and above	5

Most of the factors seemed to be based on personal judgment and there was no value that seemed to be used uniformly. However, there was some evidence obtained from a study of the survey that typical values would be 10 per cent to be added to the sensible heat load and 5 per cent to the latent load. In a few cases an over-all percentage factor was added to the total load.

SUMMARY

With the calculated cooling loads varying from 5.23 to 12.7 tons of refrigeration for the banking problem and with a resulting average of 8.33 tons, it was found impossible to classify the loads for the various cities according to any territorial grouping.

The design outside and inside temperatures used checked closely with recommended values given in THE GUIDE except for the cities mentioned.

A majority of the computations employed THE GUIDE method with variations for determining the solar radiation effects, but several rational procedures were also used which considered actual solar intensities and the nature of the surface.

The greatest variables were found in the methods adopted for estimating the amounts of uncontrolled infiltration and controlled ventilation air. The analysis indicated that a majority of load determinations considered a positive pressure in the enclosure with air introduced in the amounts of 10 to 15 cfm per person.

The results of this survey show clearly that with the different calculation methods now in use it is practically impossible for engineers even in the same locality to appraise the total cooling load on a comparable basis within close limits.

[7] Buffalo Adopts Comfort Air Conditioning Standards (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September 1936).

RESEARCH INVESTIGATIONS

Observations made in connection with this survey suggest the following research investigations to eliminate several of the variables now existent in a cooling load determination.

1. Study of methods for determining proper outside design air conditions from Weather Bureau records.
2. Continuation of basic physiological studies to determine the correct inside design air conditions for health and human comfort in various types of applications and for specified periods of occupancy.
3. Additional heat transmission tests for new building insulators, glass blocks, and metal building construction.
4. More complete data on heat absorbing glass and shading devices.
5. Additional data regarding effect of heat storage capacity of walls on the time displacement of wall conduction with respect to temperature difference and intensity of solar radiation.
6. More information with reference to effect of heat storage capacity on the magnitude of actual transient heat flow with respect to the magnitude of assumed steady-state flow.
7. Solar radiation absorption characteristics of outside building surfaces in terms of the angle of surface with respect to the rays of the sun.
8. Infiltration tests on new types of metal and wood windows.
9. Investigations of infiltration requirements for swinging and revolving doors.
10. Continuation of the studies for determining the proper ventilation requirements.
11. Initiate a program for determining heat gain from various electrical, gas and steam appliances.
12. Heating effects of artificial illumination and electrical machinery.
13. Studies to check actual cooling requirements of a building with calculated cooling load.

ACKNOWLEDGMENT

The preparation of this paper was made possible by the generous assistance of those 50 Society members who were interested in the project and willing to spend the necessary time to prepare a sample calculation. It is hoped that the needed scientific data referred to in this survey will further the investigations of more research in this field and benefit the entire profession as well as those Society members who have contributed data from their personal experience.

DISCUSSION

H. F. HUTZEL (WRITTEN): As pointed out in this paper there is a great need for unification or standardization of cooling load calculations. Probably the greatest discrepancy lies in determining the sun radiation component of the total load. Doubtless, many of the calculators figured the sun load to apply on both the south and west walls and on the roof. Since the radiation components on these three surfaces are not in phase relationship there is little justification for considering these three surfaces exposed to the sun unless the proper sun effect coefficient applying to these three surfaces is used at one particular time of the day.

Since this happened to be a banking space which would have short working hours, the maximum sun intensity on the west wall would not apply until after the bank was closed for the day. In the calculations which I submitted, the radiation load for both the roof, west and south walls was figured, but only the peak sun load or combination of sun loads which existed at one time were applied. The other surfaces which may be at the same time of the day exposed to the sun were figured for normal heat transmission.

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In analyzing Table 6, and more specifically in columns, 25, 26 and 27, it will be noted that quite a variance in amounts of air due to uncontrolled infiltration existed. This would naturally be expected since the specification only stipulated the occupancy and did not indicate the probable hourly patronage, which obviously has a direct bearing on the amount of air introduced through revolving and swing doors. In the sample calculation given in the paper as Table 1, sun loads are indicated as applying to both the south and west walls, as well as the roof. It is my opinion, had this job been calculated for sun exposure on the roof and south wall at one time, with just a normal transmission load on the west wall, the total load calculations would have doubtless been nearer in accord with the actual load. In my solution I disregarded the south wall sun load and also the roof sun load, but applied the sun load on the west wall alone for the reason that this happened to be the largest sun load on the three surfaces at any one time during the day. Probably my calculations would have been nearer correct had the roof and south wall been considered as exposed to the sun.

C. P. LICHTY (WRITTEN): The results shown in Table 6 indicate better results than one would anticipate. With the present demands for comfort cooling there has been little time for older and experienced engineers to thoroughly become familiar with cooling load computations and, in addition, the younger engineers, who have become associated with the business recently, have not had time to gain experience for applying sound judgment to practical problems. It seems to me that this paper indicates a definite step of progress by the Society in establishing more scientific methods for future cooling load computations.

A. L. MAILLARD (WRITTEN): I am not familiar with the individuals who supplied the data for Kansas City in these computations, but it seems as though they agree closely enough on their results to be typical of everyday experience. I was somewhat surprised to notice how great discrepancies occurred in figures submitted by different people in the same localities. It is very evident that in these cases some standardization would be desirable. In this connection the Air Conditioning Council in Kansas City has been functioning for the past four years. The Council has established minimum standards with the result that it is seldom there is a variance of 10 per cent in the load calculations for any installation.

No. 1064

CALCULATED OVER-ALL COEFFICIENTS FOR WALLS WITH AIR SPACE INSULATION

By F. B. Rowley* (*MEMBER*), Minneapolis, Minn.

THE over-all coefficient of heat conductivity U for a wall can be readily calculated if the thermal properties for the materials from which the wall is built are known, and if these materials are all placed uniformly over the full area of the wall in such manner that all of the heat must pass through all of the uniform layers of material in series. In this case the over-all heat resistance of the wall is equal to the sum of the heat resistances of the various materials and is uniform for all areas of the wall.

If the materials are not of uniform thickness over the full area of the wall as is the case when insulating materials are placed between the studs of a frame wall or in the cores of hollow concrete block or tile walls, then all of the heat passing through the wall does not pass through equal thickness of all materials and the heat resistance is not the same for all areas of the wall. In these cases there are parallel paths of heat flow and the heat resistance may be entirely different for the various paths. In some instances the heat resistance for the parallel paths are so nearly equal, or other conditions in the wall are such that all sections of it may be considered as having uniform heat resistance and the simple method of adding heat resistance to determine the over-all coefficients may be followed. There are, however, many combinations of materials which should be used with caution when an insulated wall is required, and in some instances present practice should be somewhat modified in order to get a true picture of the insulating value of material as applied. To illustrate these principles a few examples will be discussed, and in discussing them the standard coefficient of conductivity and conductance for various sections of the wall will be used as given in The A.S.H.V.E. Guide. Briefly it should be remembered that the reciprocal of any conductivity or conductance coefficient for a specific part of the wall represents the heat resistance of that particular part of the wall, and that if the materials are placed in uniform layers so that all of the heat must pass through them in series, the summation of all resistances represents the over-all resistance of the complete wall section. Thus the resistance of individual parts of the wall through which the heat flows in series may be added and the reciprocal of this summation will be equal to the over-all coefficient of conductivity for the complete wall. A more complete discussion of the problem may be found in The Guide.

To illustrate the principle involved consider the four walls, Nos. 1, 2, 3 and

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Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Swampscott, Mass., June, 1937, by C. M. F. Peterson.

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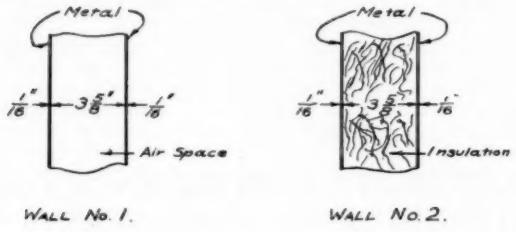
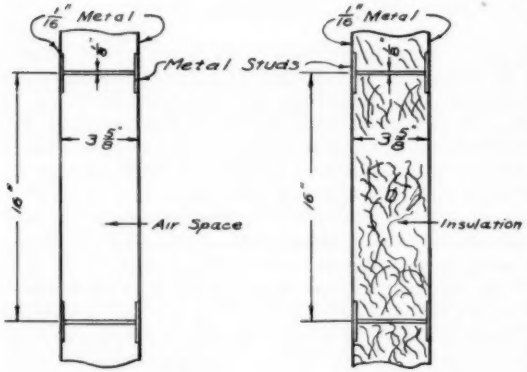


WALL No. 1. WALL No. 2.

FIG. 1. TYPICAL METAL FRAME WALL CONSTRUCTION



WALL No. 3 WALL No. 4

4 of Fig. 1, each made of 1/16-in. thick steel plates spaced 3 5/8 in. apart and no additional surface finishes. In Wall No. 1 there are no supports between the surfaces which would form a path for heat flow. Wall No. 2 is the same as Wall No. 1 with the exception that the air space between the plates is filled with a homogeneous insulating material. In Wall No. 3 there are 1/8-in. thick steel studs spaced 16 in. on center and securely bonded by a weld to each of the steel surfaces. Wall No. 4 is the same as No. 3 with the exception that insulating material has been added between the studs. For the purpose of calculating the over-all coefficients of conductivity U for the various walls consider the coefficients of the different materials as follows:

Surface conductance, f_1 and $f_0 = 1.65$
Thermal conductivity k for steel $= 420$
Air space conductance, 1 in. or more in thickness $= 1.1$
Thermal conductivity k of insulating material $= 0.27$
Thermal conductivity k of wood (to be used in Walls Nos. 5 and 6) $= 0.8$.

In the four walls under consideration the heat resistance is the same for the exterior and interior surfaces and also for the steel plates forming two sides of the wall. The resistance of the steel plate is, however, so small in comparison with other parts of the wall that it may be neglected in making calculations. In Walls Nos. 1 and 2 all of the heat flowing through the central section of the wall passes through paths of equal resistance. For Walls Nos. 3 and 4 the heat flows in parallel paths of very different heat resistances and the calculations for these walls must be made on an entirely different basis.

METAL FRAME WALL CONSTRUCTION

In calculating the over-all coefficient for the four walls the resistance for the surfaces and everything excepting the central section may be taken as equal for all walls, and since the material itself has such a low resistance the total resistance for this part of all walls will be equal to that of two surface coefficients, or $2/1.65 = 1.21$. The total heat resistance for Wall No. 1 becomes $1.21 + \frac{1}{1.1} = 2.12$ and $U = 1/2.12 = 0.472$. For Wall No. 2 the resistance becomes $1.21 + \frac{3.625}{0.27} = 14.61$ and $U = \frac{1}{14.61} = 0.069$. In this application the insulating material has covered the full area of the wall and the full heat resistance of the material is introduced into all paths for heat flow through the wall, and the resistances of all materials may be properly added to obtain the over-all heat resistance of the wall.

For Walls Nos. 3 and 4 the conditions are entirely different; there are parallel paths of different thermal conductivities through which the heat may flow. Since the steel surfaces of the walls are themselves good conductors, the heat will be free to flow laterally through these surfaces and feed any paths of heat flow through the wall to its full conducting capacity, and there will be substantially parallel flow at different rates through the various sections.

If parallel flow of heat is accepted for walls of this type, the average conductivity for the central section may be calculated and the reciprocal of this taken as the average resistance of that part of the wall. The calculations are as follows:

For Wall No. 3:

$$\text{Average conductance} = \frac{\left(\frac{420}{3.625} \times 0.125\right) + \left(1.1 \times 15.875\right)}{16} = 2.00$$

and R for the complete wall $= 1.21 + \frac{1}{2.00} = 1.70$; $\quad U = \frac{1}{1.70} = 0.59$

For Wall No. 4:

$$\text{Average conductance} = \frac{\left(\frac{420}{3.625} \times 0.125\right) + \left(\frac{0.27}{3.625} \times 15.875\right)}{16} = 0.98$$

and R for the complete wall $= 1.21 + \frac{1}{0.98} = 2.23$; $\quad U = \frac{1}{2.23} = 0.45$.

Table 1 shows the calculated over-all coefficients for the walls with and without insulation applied, and, while it may not be strictly correct to assume parallel heat flow through the materials making up the central sections of Walls Nos. 3 and 4, this is substantially the condition which would exist when there are high conductivity materials forming the two surfaces of the walls, and shows a condition often overlooked in the application of insulating mate-

TABLE 1—CALCULATED COEFFICIENTS FOR WALLS NOS. 1, 2, 3 AND 4

WALL NO.	1	2	3	4
Over-all conductance.....	0.47	0.069	0.59	0.45

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rials. If the surface materials were of low conductivity there would still be a high rate of heat conductance through the steel studs but the problem would be more complicated to calculate and any insulating material between the studs would be more effective. There would, however, be cold spots or lines on the inside surface of the wall over the studs in cold weather. The necessity of spreading insulating material over the full area of the wall and eliminating any path of high conductivity through insulated sections of a wall was discussed in a previous paper.[1] The ultimate effect of any paths of high conductivity will depend upon their area in comparison to the total area of the wall and upon the ease with which heat may be fed to and taken away from these paths. For instance, if the materials making up the inner and outer surface finishes of the wall are of high conductivity, then any path through the wall of high conductivity will do much more damage than if the surfaces are made up of some material which has a high heat resistance and will not conduct the heat laterally to channels where it may easily pass through the wall.

WOOD FRAME WALL CONSTRUCTION

A condition met extensively in practice is shown by Walls Nos. 5 and 6 in Fig. 2. These may be considered as typical frame construction using 2x4 in. studs spaced 16 in. on center with an inside surface finish composed of wood lath and plaster, and an outside surface finish of fir sheathing, building paper and lap siding. Wall No. 6 is insulated with a fill material having an assumed conductivity coefficient *k of* 0.27 per inch of thickness. The surfaces of these walls are not of high conductivity material, yet the plaster on the inside surface does have a rather high conductivity when compared with the other materials in the wall. It has become common practice with walls of this type to add insulating materials between the studs of the wall and to credit the insulating value of

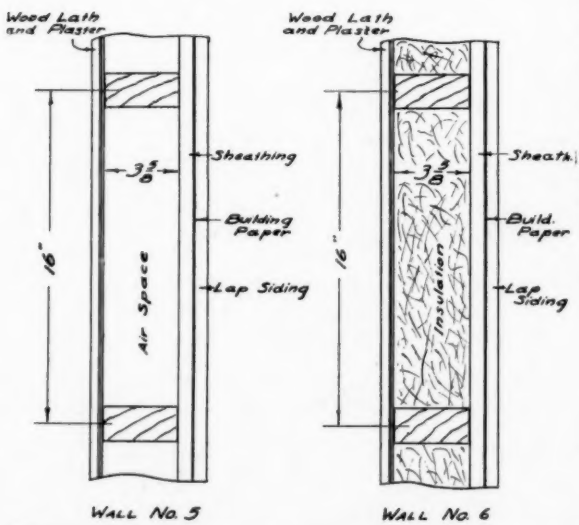

FIG. 2. TYPICAL WOOD FRAME WALL CONSTRUCTION

[1] Thermal Properties of Concrete Construction, by F. B. Rowley, A. B. Algren and Clifford Carlson, A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936.

this material to the entire surface of the wall. Since the wood studs are in themselves fairly good insulators this method of calculation does not lead to the same error that it would for a steel wall with steel studs, but nevertheless it does lead to an error and repeated tests on frame walls insulated in this manner have shown that the over-all coefficients as determined by tests are higher than those calculated for the wall when assuming that the insulating material covers the full area of the wall. If, however, the calculations for the central section of the wall are made on the basis of parallel flow through the insulating material and the studs, the calculated results check very closely with test results.

On the basis of parallel flow the average conductance coefficient for the central section of Wall No. 5 when calculated is,

$$\frac{\left(\frac{0.8}{3.625}\times 1.75\right)+\left(1.1\times 14.25\right)}{16}=1.003.$$

Many tests have shown that the conductances of the central section of walls of this type where the air space is over 1 in. in thickness is substantially the same whether the studs are present or not. This is probably due to the fact that there is a transfer of heat between the surfaces of the studs bounding the air space and the air in the space and 1.1 may be considered as a fair value to assign to this space for a wall constructed as No. 5.

Fill Insulation

For Wall No. 6 the conductance coefficient for the central section is,

$$\frac{\left(\frac{0.8}{3.625}\times 1.75\right)+\left(\frac{0.27}{3.625}\times 14.25\right)}{16}=0.0905.$$

The total heat resistance which should be credited to material added between the studs is the difference in heat resistance over the central section of the wall between the insulated and non-insulated walls. In terms of heat resistance, this becomes

$$\frac{1}{0.0905}-\frac{1}{1.003}=10.05$$

as the actual heat resistance added to the wall, whereas by the method which has been in common use, of crediting the insulation with the full area of the wall, the added resistance would have been

$$\frac{3.625}{0.27}-\frac{1}{1.1}=12.52$$

a difference of $12.52 - 10.05 = 2.47$.

This difference becomes an appreciable factor when considering walls with heavy insulation between the studs. In the particular example quoted, Wall

No. 5 without insulation and with a 15 mph wind velocity over the outside surface has a commonly accepted over-all coefficient $U = 0.25$. If in the addition of insulation between the studs having a coefficient of 0.27, calculations are made in the commonly accepted manner giving the insulation credit for the full area of the wall and deducting the value of an average air space, the final heat resistance will be,

$$R = \frac{1}{0.25} - \frac{1}{1.1} + \frac{3.625}{0.27} = 16.52 \text{ and } U = \frac{1}{16.52} = 0.061.$$

If the calculations for the insulated wall are made on the basis of parallel flow and insulation is credited for only that part of the wall actually covered by the insulation, the total heat resistance of the wall becomes,

$$R = \frac{1}{0.25} + 10.05 = 14.05, \text{ and } U = \frac{1}{14.05} = 0.072.$$

Conductivity tests by the hot box method for frame walls insulated as in Wall No. 6 show a conductivity coefficient somewhat higher than that calculated by the method of parallel heat flow, indicating that the studs are effective in carrying a somewhat higher proportion of the heat than is credited to them by this method of calculation. The difference, however, is not great and probably 0.075 would be an average over-all test coefficient for such a wall.

Reflective Insulation

A similar practice has been followed in making calculations for reflective insulation placed between the studs of frame walls. The insulation has been credited with covering the full area and not confined to that part between the studs. The same condition exists for this type of insulation as for the fill insulation, and the same precautions must be taken in making calculations. As in the case of fill insulation the errors may be very great for walls built of high conductivity material.

SUMMARY

A brief analysis of the revised over-all coefficients of heat conductivity values given in Chapter 5 of THE GUIDE 1937 shows that the figures are more precise where insulation covers only a portion of the frame wall construction. Wall No. 41-I in THE GUIDE 1936 insulated with rockwool fill between the studs had an over-all coefficient of 0.061, while the coefficient for the corresponding wall, No. 65-I in the 1937 edition is 0.072. Similar revisions are found in the coefficients for walls with reflective types of insulation used between studs. For example, Wall No. 41-H in THE GUIDE 1936 with bright aluminum foil as a lining for one side of the air space between the studs had a coefficient of 0.19 and the corresponding wall No. 65-H in THE GUIDE 1937 has a coefficient of 0.20.

In most cases the new coefficients in THE GUIDE show relatively small changes, although in comparing the coefficients on a percentage basis they become more important.

No. 1065

RADIATION AND CONVECTION ACROSS AIR SPACES IN FRAME CONSTRUCTION

By GORDON B. WILKES * AND CARL M. F. PETERSON ** (*MEMBERS*)
CAMBRIDGE, MASS.

THE fact that the air space between the studs in ordinary frame construction had definite insulating value has been long known, but its actual value was not appreciated until comparatively recent years.

In 1929 Prof. F. B. Rowley [1] was one of the first to attempt to evaluate the actual insulating effect of ordinary air spaces, and subsequently considerable interest was aroused in air space insulation when the U. S. Bureau of Standards, in Circular No. 376, stated that from 50 to 80 per cent of the heat transfer across ordinary air spaces was due to radiation. A number of other experimenters, including Schad, Queer, Van Dusen, Gregg, Mason, Griffiths and Davis, determined the insulating value of air spaces, but none of these, with the exception of Griffiths and Davis, attempted to evaluate separately radiation and convection. It was also just about this time that metal foil for insulation became commercially available so that one could minimize the radiant transfer of heat across an air space. The principles involved in air space insulation are radically different from those for ordinary insulators and there has been much confusion among engineers as well as laymen due to a lack of knowledge of the fundamental laws of radiation and convection as well as a lack of experimental data.

At the present time there are practically no data available in regard to the rate of heat transfer across air spaces that are not in the vertical position, although it is well known that the rate of heat transfer across horizontal air space must be much greater when the heat is passing upward rather than downward. This difference is much more marked in the case of reflective type insulation, because radiation is minimized and convection remains the chief factor involved in the heat transfer.

The purpose of this paper is two-fold: first, to state again the fundamental laws of radiation and convection because of the apparent lack of understanding;

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* Professor, Mechanical Engineering, Massachusetts Institute of Technology.
** Instructor, Mechanical Engineering, Massachusetts Institute of Technology.

[1] Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren, A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929.
Insulating Value of Bright Metallic Surfaces, by F. B. Rowley, A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1937, by G. B. Wilkes.

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and second, to give new experimental data in regard to the rate of heat transfer across full-sized air spaces in various positions bounded by surfaces that are both good and poor reflectors of heat radiation.

Radiation

The fundamental laws of radiation are well established, both theoretically and experimentally. The transfer of heat by means of radiation across an air space is dependent upon the temperature of the two boundary surfaces and their respective emissivities. Thickness of the air space has no appreciable effect upon the amount of radiation transfer. For example, if two infinite

TABLE 1—TOTAL EMISSIVITY OF PLANE SURFACES AT 70 F

SURFACE	TOTAL EMISSIVITY
Wood	0.9
Plaster	0.93
Paper	0.9
Black paint	0.9
White paint	0.9
Aluminum paint	0.4 –0.65
Coated papers	0.32–0.64
Aluminum foil	0.05
Aluminum oxidized	0.11
Copper polished	0.04
Copper oxidized black	0.78
Iron polished	0.06
Iron, rough oxide layer	0.81
Glass	0.9
Chrome plate, polished	0.04

parallel plates are considered at temperatures of 60 F and 20 F, respectively, the rate of heat transfer by radiation would be essentially the same if the distance between these plates were one inch, one foot, or one mile.

The emissivity of a plane surface is merely a factor indicating the relative amount of radiation from the surface in question as compared to a perfect radiator (*black body*) under the same conditions. If the emissivity of a *black body* is termed unity, then the emissivity of all real surfaces will be less than one. The emissivity of a number of common surfaces found in buildings at a temperature around 70 F is given in Table 1.

From the values given in Table 1 it will be noted that most of the non-metals have high emissivities in the neighborhood of 0.9, while the metals when not oxidized have very low emissivities around 0.05. Aluminum and bronze paint that are a combination of metals and a non-metallic vehicle have an emissivity about half-way between the two values stated.

Heat radiation striking an opaque surface must either be absorbed or reflected. Consequently, the radiation absorbed by a surface plus that reflected must equal in amount the radiation striking the surface. It is also obvious that if a material is a good absorber of radiation, it must be also an equally good emitter of radiation. For example, if a body is placed in an evacuated

space and allowed to come to temperature equilibrium, the radiation absorbed must equal that emitted.

Consequently, if a = the absorptivity, e = the emissivity, and r = the reflectivity of a surface, then

$$a = e = 1\text{-}r \tag{1}$$

As far as the rate of heat transfer across an air space by radiation is concerned, it makes no difference whether the reflective surface is on the cooler or warmer side of the space.

Emissivity varies with the temperature of the surface, while absorptivity may vary with the temperature of the source of radiation as well as the temperature of the receiving surface. Black and white paint have essentially

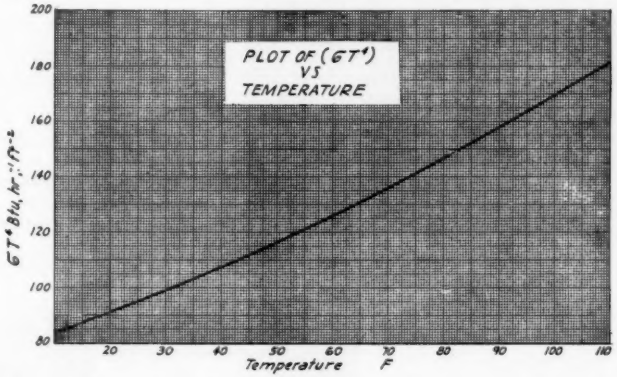

FIG. 1. CURVE SHOWING PLOT OF σT^4 VS. TEMPERATURE, DEGREES FAHRENHEIT

the same emissivity at room temperature, but if exposed to solar radiation one will find that the black paint will absorb many times as much heat as the white paint. Aluminum foil will emit at room temperature only about 5 per cent as much radiation as a white painted surface but, when exposed to radiation from the sun, the aluminum will not reflect as well as the white paint.

Every surface, regardless of its temperature, is radiating and receiving heat by radiation. In the case of the transfer of heat by radiation across an air space, one is concerned only with the net heat transfer between the two surfaces at different temperatures. This can be calculated by the well-established Stefan-Boltzmann Law.

$$q_r = A\, e_{eff}\, \sigma\, (T_1^4 - T_2^4) \tag{2}$$

where

q_r = net rate heat radiation, Btu, hour.
A = area, sq ft.
e_{eff} = effective emissivity.
σ = Stefan's constant = 17.2×10^{-10} Btu, hour^{-1}, foot^{-2}, $F_{abs.}^{-4}$
T_1 = absolute temperature deg F $(460 + t_1)$ of warmer surface.
T_2 = absolute temperature deg F $(460 + t_2)$ of cooler surface.

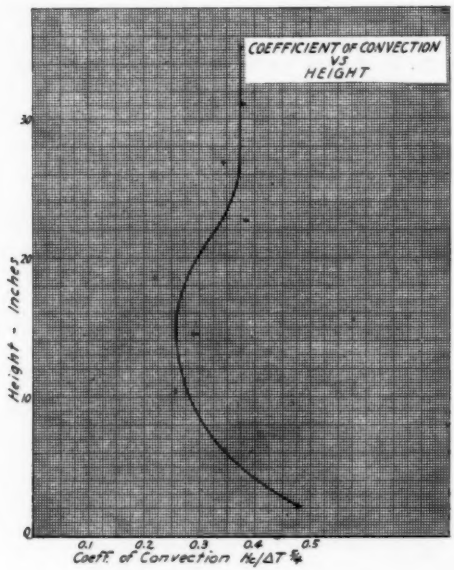

FIG. 2. CURVES INDICATING VARIATION IN COEFFICIENT OF CONVECTION VS. HEIGHT

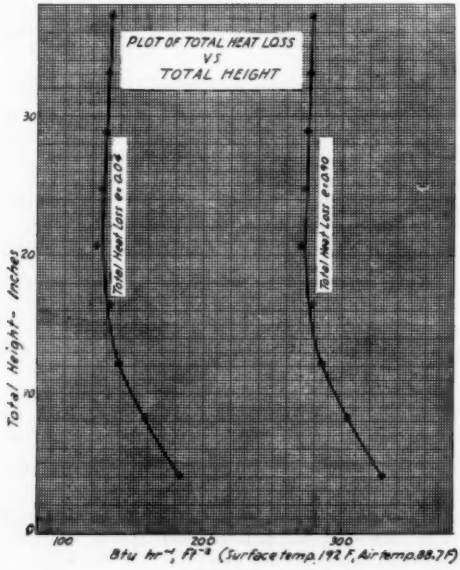

FIG. 3. CURVES INDICATING TOTAL HEAT LOSS VS. TOTAL HEIGHT

In dealing with the net radiation between two parallel surfaces, the emissivity of each surface must be considered. It can be demonstrated that the *effective* emissivity or e_{ett} used in equation 2 is a combination of the two individual emissivities. This effective emissivity can be calculated as follows:

$$e_{ett} = \frac{1}{\frac{1}{e_1} + \frac{1}{e_2} - 1} \qquad (3)$$

where

e_1 = emissivity of warmer surface.
e_2 = emissivity of cooler surface.
For example, if $e_1 = 0.90$ and $e_2 = 0.90$; then $e_{ett} = 0.82$.
and, if $e_1 = 0.05$ and $e_2 = 0.90$; then $e_{ett} = 0.05$.

Formula 3 is based on infinite parallel plates, but there will be little error involved if it is applied to the ordinary air spaces found in frame construction.

To make easier the calculation of the transfer of heat by radiation, Fig. 1 gives values of σT^4 for the usual temperatures of the surfaces facing such air spaces.

Example: Find the heat transfer by radiation across an air space between studs, where one surface is paper (plasterboard) at a temperature of 60 F and the other wood sheathing at a temperature of 20 F.

From chart $\sigma T_1^4 = 126.0$ Btu per hour per square foot.
$\sigma T_2^4 = 91.2$ Btu per hour per square foot.

Difference $\sigma (T_1^4 - T_2^4) = 34.8$ Btu per hour per square foot.

$$\text{The effective emissivity} = \frac{1}{\frac{1}{0.9} + \frac{1}{0.9} - 1} = 0.83.$$

For each square foot of surface, the amount of heat transferred by radiation will be

$$q_r = e_{ett}\ \sigma\ (T_1^4 - T_2^4) = 0.83 \times 34.8 = 28.9 \text{ Btu per hour per square foot}$$

Convection

Convection as a means of transfer of heat across an air space follows very different laws and is much more complicated than radiation or conduction.

It has been demonstrated experimentally by a number of experimenters that heat transfer by natural or free convection varies approximately with the 5/4 power of the temperature difference between the surfaces on each side of

TABLE 2—TOTAL HEAT LOSS PERCENTAGE ERRORS FOR VARIOUS HEIGHTS

HEIGHT OF TEST SPECIMEN, IN.	TOTAL HEAT LOSS BTU PER HOUR PER SQ FT	PER CENT ERROR
6	170	31.0 high
9	154	18.5 high
12	140	8.0 high
>16	130	Correct

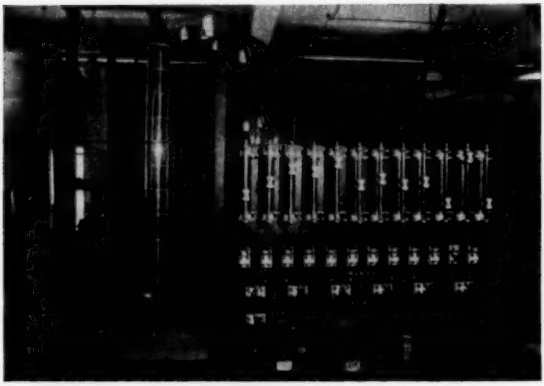

FIG. 4. VIEW OF APPARATUS USED TO DETERMINE VARIATION OF CONVECTION WITH HEIGHT

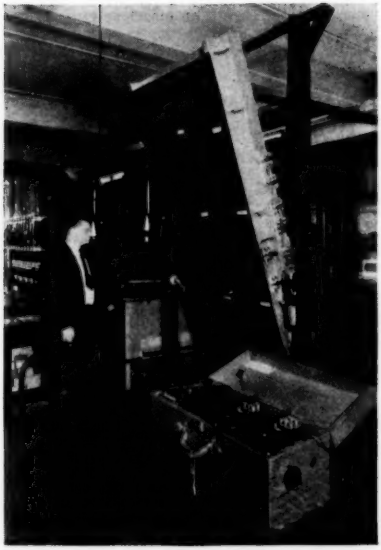

FIG. 5. VIEW OF AIR SPACE APPARATUS

an air space. For temperature drops as usually found in building walls, the transfer of heat by both conduction and radiation is approximately proportional to the temperature difference. If the temperature difference across an air space were increased from 5 F to 40 F, the radiation would be increased about eight times, while the convection would be increased more than twelve times.

In the case of radiation, the position of the air space (vertical, horizontal, etc.) has no effect, but it has a very marked effect on convection. The rate of heat transfer by convection across a horizontal air space with the heat flowing upward is roughly three times what it would be with the heat flowing downward.

Convection also varies with the height of the air space up to approximately 2 ft. Figs. 2 and 3 indicate this variation as found in an apparatus set up in

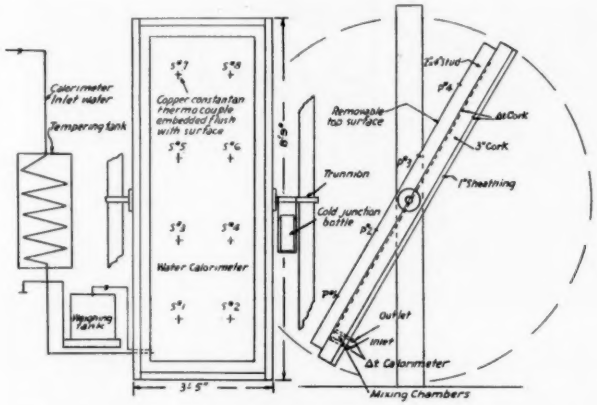

FIG. 6. DIAGRAMMATIC ARRANGEMENT OF AIR SPACE APPARATUS

the laboratory, as shown in the photograph of Fig. 4, and which gives essentially the same data as found by Griffiths and Davis. It will be seen readily that experiments upon natural convection must be made upon nearly full size air spaces, and that tests made upon samples 1 ft high will give higher values if applied to an 8 ft air space.

Since there is apparently little information available in regard to the transfer of heat across air spaces other than in the vertical position and since most of the work on vertical air spaces has been done on relatively small specimens, with little attempt made to separate radiation from convection, it seemed important to carry out an investigation on full sized air spaces. Furthermore, Table 2, given in chapter 5 of THE GUIDE 1937 for conductances of air spaces assumes that both radiation and convection vary proportionately with the temperature difference and also with the mean temperature. This assumption is approximately true for radiation, but is far from true for convection. With ordinary surfaces facing a vertical air space, such as plaster, wood, paper, etc., radiation is responsible for roughly 65 per cent of the total heat transfer and

the error is not very serious. In the case of the newer type of reflective insulation, the transfer of heat by radiation is minimized, thus making convection responsible for most of the heat transfer. In this latter case the assumptions made above will lead to very serious errors in the calculation of the insulating value.

It was decided to construct an air space 8 ft high by 2 ft-8 in. wide and 3⅝ in. across, as shown in the photograph of Fig. 5 and diagrammatically in

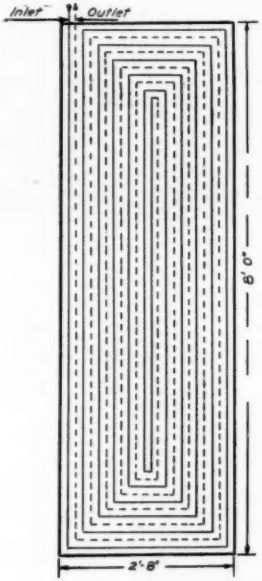

FIG. 7. DIAGRAM SHOWING COPPER TUBING ARRANGEMENT IN WATER CALORIMETER

Fig. 6. This would correspond to about the normal height of a frame wall air space.

One side of this air space consisted of a steel plate with 180 ft of ¼ in. copper tubing attached as shown in Fig. 7 to make a large water calorimeter. This was backed with 3 in. of corkboard so as to reduce the gain or loss of heat from the room. A frame was built of 2 x 4 in. studs around this plate and on these were fastened either black building paper or aluminum foil on paper, thus enclosing the air space and providing means of minimizing the radiation when desired.

Eight thermocouples were cemented in grooves cut in the steel plate so that its average temperature could be accurately measured. Thermocouples were

also attached to the paper or foil on the other side of the air space. In this way the temperature drop across the air space could be determined with precision.

The inlet and outlet temperature of the water in the calorimeter was measured by means of multiple thermocouples in a suitable mixing chamber close to the calorimeter. Before entering the calorimeter, the water passed through a copper coil immersed in water to maintain a constant temperature. This temperature could be varied at will by either heating or cooling the water in the tempering tank. The arrangement of the copper coil is given in Fig. 7, and it produced very uniform temperatures over the whole area of the calorimeter plate.

The whole apparatus was mounted on trunnions so that it could be tipped

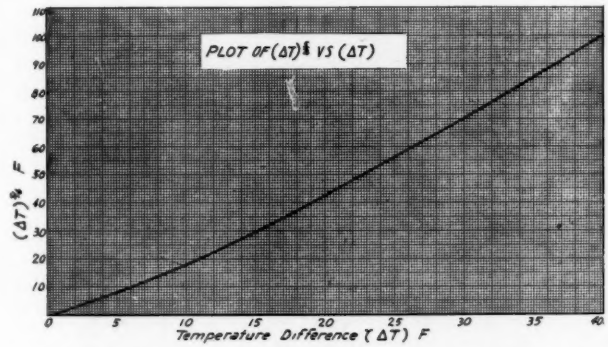

FIG. 8. CURVE SHOWING PLOT OF $\Delta T^{5/4}$ VS. TEMPERATURE DIFFERENCE, DEGREES FAHRENHEIT

at any angle desired and with the heat either flowing upward or downward when it was in any position other than vertical.

A semi-precision potentiometer was used to measure the electromotive force developed by the thermocouples. It was possible to read to 0.001 millivolt, which corresponded to about 0.045 F on the single thermocouples and about 0.009 F on the multiple couples for the calorimeter.

Thermocouples were placed on each surface of the cork insulation so that a suitable correction could be made for any gain or loss of heat. The edge loss of heat through the 2 x 4 in. frame was small and is included in the loss through the cork back.

All cold junctions of the thermocouples were kept in thermos bottles filled with cracked ice and water. A careful calibration was made before starting the work.

Tests were conducted in five different positions, horizontal, and 45 deg with heat flowing downward, horizontal, and 45 deg with heat flowing upward, and in the vertical position. Tests were made with aluminum foil facing one side of the air space or with black building paper. In order to determine the

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coefficients of convection for the different positions, aluminum foil was used for the outer surface so as to minimize radiation. The temperature difference was varied and in one case the direction of heat flow was reversed by sending heated water through the calorimeter rather than cool water. The rate of heat transfer was not affected by having the foil surface absorbing rather than emitting heat.

RESULTS

The accompanying curves express the results of these experiments much better than a table of data.

The relation between temperature difference, degrees Fahrenheit, and the temperature difference raised to the ¾ power, for use in computing convection,

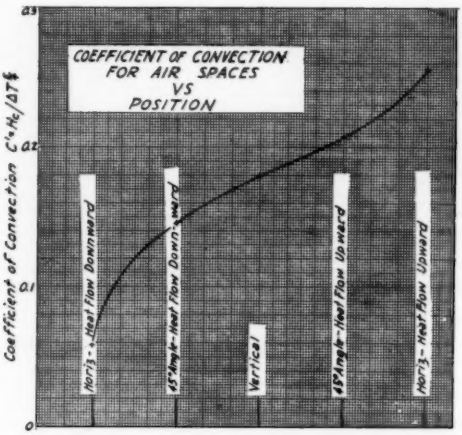

FIG. 9. CURVE INDICATING COEFFICIENT OF CONVECTION FOR AIR SPACES VS. POSITION

is shown in Fig. 8. It is interesting to observe that if one has 40 F temperature difference rather than 10 F, the heat transfer by convection will be increased 5.6 times rather than just 4 times, as would be the case if convection varied with the temperature difference directly. Radiation, on the other hand, for this same range of temperature would increase nearly proportional to the temperature difference, or about 4 times. For example, if one had a conductance value (radiation plus convection) for a foil-lined air space that was experimentally determined, with a temperature difference of 10 F, and it were applied to a case with a 40 F temperature difference, by merely multiplying the conductance value by the temperature difference, the new value would be roughly 25 per cent too low. This is the method in common use today.

Experimental values for the variation of the coefficient of convection with height are indicated in Fig. 2. The coefficient is greatest at small heights,

gradually diminishing to a minimum value at roughly 15 in. and then becoming uniform, regardless of height, above 24 in. This effect is probably due to streamline motion of the air over the surface up to a height of about 15 in. and then the air becoming turbulent causes the coefficient to remain constant for all heights greater than about 2 ft. The smoke from a lighted cigarette placed close to the bottom of the column demonstrated clearly the change from streamline to turbulent motion of the air stream.

The variation of the total heat loss versus the total height is indicated in Fig. 3. It shows clearly the error that one can make if data are taken from a relatively small area of low height and then applied to greater heights. With a chrome-plated sample ($e = 0.04$) and temperature conditions corresponding to

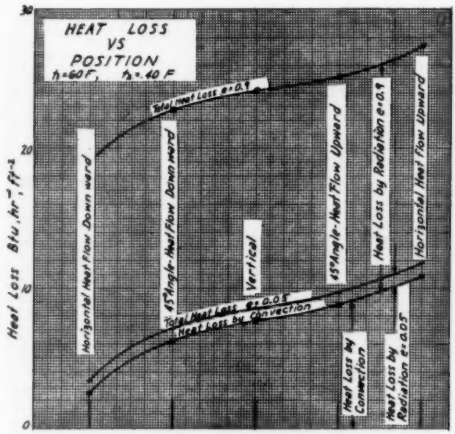

FIG. 10. CURVES SHOWING RELATION OF HEAT LOSS VS. POSITION

those indicated in Fig. 3, the values given in Table 2 will show the error to be expected when experimental data are taken for different heights and then applied to heights greater than 16 in.

In Fig. 9 the effect of position on the coefficient of convection for air spaces is shown. The usual coefficient of convection is designated by C and deals with the transfer of heat from a surface to the surrounding air. The coefficient used in this chart deals with the transfer of heat by convection from one surface to another parallel surface and will be designated by C'. In other words, the temperature difference between two parallel surfaces is used in calculating C' rather than the temperature difference between a surface and the surrounding air that would be used in the case of C.

The large effect that position plays in convection is shown clearly. In the horizontal position, if heat is flowing upward, the coefficient is 0.256 as compared with a value of 0.058 with the heat flow downward, or roughly five times

as great. The value for the vertical position is not half-way between these two values but is nearer the higher value, namely, 0.180.

The relation between heat loss and position is illustrated in Fig. 10 for an air space with and without a reflective surface. It also shows the relative amount of heat transferred by convection and radiation for both cases. The results indicated on this chart apply only when there is a 20 F difference in temperature between the surfaces and also when one surface is approximately 60 F and the other 40 F.

The values in Table 3, taken from Fig. 10, show the important part that radiation plays in the transfer of heat when the air space is bounded by

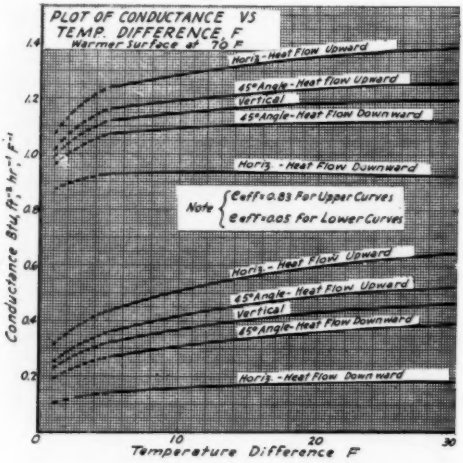

FIG. 11. CURVES OF CONDUCTANCE VS. TEMPERATURE DIFFERENCE, DEGREES FAHRENHEIT

ordinary surfaces ($e_{eff} = 0.9$) as compared to the minor part when one of the surfaces is of the reflective type ($e_{eff} = 0.05$).

The variation of conductance with temperature differences that might be found in a building wall air space is shown in Fig. 11. The calculation of radiation is based on the warmer surface being at 70 F. The variation due to mean temperature is not great if it is not more than one would find in building walls, especially in the case where one surface is of the reflective type. In this latter case the error, using data from the chart, would only be from 1 to 3.5 per cent too low if the warmer surface were 90 F and from 1 to 3.5 per cent too high if the warmer surface were 50 F instead of 70 F. This is, of course, due to the small part that radiation plays in the heat transfer when a reflective surface is used. In the case of ordinary surfaces with effective e equal to 0.83, the error would be somewhat greater. With the warmer surface

at 90 F, the values from the chart would be from 6.7 to 10 per cent too low and with the warmer surface at 50 F instead of 70 F the values would be from 5.9 to 8.7 per cent too high.

If one knows the temperature drop across an air space, with or without one reflective surface, Fig. 11 can be used to determine the total heat transfer across the air space in various positions, provided the temperature ranges are not too far from those used in calculating the chart.

Frequently the insulating value of an air space is compared to that of a good average insulating material. If a temperature difference of 20 F is assumed for a good average insulating material having a coefficient of thermal conductivity equal to 0.30 Btu per hour per square foot per inch per degree

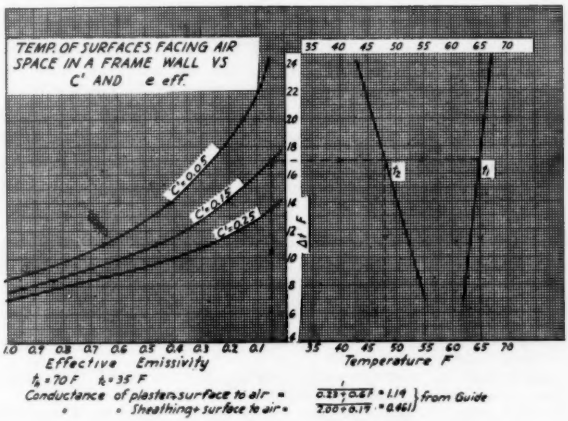

FIG. 12. CHART SHOWING TEMPERATURE OF SURFACES FACING AIR SPACE IN A FRAME WALL VS C' AND e_{eff}

Fahrenheit, Table 4 will give the equivalent thickness of such a material corresponding to various types of air spaces.

The values in Table 4 illustrate the large difference in insulating value of an air space with one reflective surface due to its position. It is more than three times as good an insulator in a horizontal position with the heat flowing downward as when in the same position with the heat flowing upward.

The method of calculating the radiation and convection of heat across an air space has been shown, provided the temperatures of the bounding surfaces are known. This is fundamental if one is to design air space insulation intelligently. The architect or builder, however, would have difficulty in calculating the heat loss through a wall containing an air space with just the foregoing information, as the temperatures of the surfaces facing the air space are not known accurately.

It is possible that a complicated nomographic chart could be constructed so

TABLE 3—COMPARISON OF RADIATION EFFECTS BOUNDED BY VARIOUS SURFACES

POSITION	TOTAL HEAT LOSS		PER CENT RADIATION	
	$e_{eff} = 0.9$	$e_{eff} = 0.05$	$e_{eff} = 0.9$	$e_{eff} = 0.05$
Horizontal, heat flow downward	19.0	3.4	87	26
45 deg heat flow downward	22.7	7.1	73	13
Vertical	24.2	8.6	68	10
45 deg heat flow upward	25.3	9.7	65	9
Horizontal, heat flow upward	27.4	11.8	60	8

that these temperatures could be determined for various indoor and outdoor temperatures, various wall constructions, various effective emissivities and for different coefficients of convection. In order to simplify the problem by specifying certain common conditions, assume an indoor temperature of 70 F and outdoor temperature of 35 F and a typical frame wall consisting of clapboards, building paper, sheathing, studs and plaster. With these assumptions a chart can be constructed which will readily give the necessary surface temperatures for any given effective emissivity and for any given coefficient of convection.

Assume

q = heat flow, Btu per hour per square foot.
C_1 = conductance of plaster plus surface to inside air.
C_2 = conductance of sheathing and clapboards plus surface to outside air.
t_h = temperature of inside air.
t_c = temperature of outside air.
t_1 and T_1 = temperature of warmer surface facing the air space in degrees Fahrenheit and degrees Fahrenheit absolute.
t_2 and T_2 = temperature of cooler surface facing the air space in degrees Fahrenheit and degrees Fahrenheit absolute.
C' = convection coefficient between surfaces.
e_{eff} = effective emissivity of surfaces.
σ = Stefan's constant 17.2×10^{-10} Btu, hour^{-1}, foot^{-2}, F$^{-4}_{abs.}$

Then

$$q = C_1 (t_h - t_1) = C_2 (t_2 - t_c) = C' (t_1 - t_2)^{5/4} + e_{eff}\, \sigma (T_1^4 - T_2^4) \qquad (4)$$

For the conditions assumed, THE A.S.H.V.E. GUIDE, 1937, gives values of

TABLE 4—EQUIVALENT THICKNESS OF INSULATING MATERIAL

POSITION	ORDINARY SURFACES e_{eff} 0.83	ONE REFLECTING SURFACE e_{eff} 0.05
Horizontal, heat flow downward	0.32 in.	1.67 in.
45 deg heat flow downward	0.27 in.	0.83 in.
Vertical	0.25 in.	0.70 in.
45 deg heat flow upward	0.24 in.	0.61 in.
Horizontal, heat flow upward	0.22 in.	0.50 in.

$C_1 = 1.19$ and $C_2 = 0.461$ and t_1 and t_2 can be calculated for any given e_{eff} and C'.

It is somewhat easier to assume a value of t_1 such as 67 F, then

$$t_2 = \frac{C_1 (t_h - t_1)}{C_2} + t_c = 43 \text{ F.}$$

Knowing t_1 and t_2 and assuming a value of $C' = 0.05$, one can solve the equation

$$C_1 (t_h - t_1) = C' (t_1 - t_2)^{5/4} + e_{eff}\, \sigma\, (T_1^4 - T_2^4)$$

for a corresponding value of e_{eff}, which in this case equals 0.057. This calculation will give one point on the curve in Fig. 12 for $C' = 0.05$. By repeating this process with different values of t_1, the entire curve for $C' = 0.05$ can be drawn. A similar procedure will produce curves for other values of C'.

Thus, using the left hand side of Fig. 12, the temperature drop across the air space for any given effective emissivity and for any convection coefficient may be determined. By use of the right hand side of the chart, this temperature drop can be converted into the actual temperatures of the two surfaces facing the air space. Once these two temperatures are known, the heat flow can be calculated.

Similar charts may be constructed for any type of wall and for any temperature conditions as to the inside and outdoor air. The values obtained from the chart in Fig. 12 may be used with only a small error for brick veneer walls, and therefore they can be used for most of the common types of residence construction.

DISCUSSION

PROF. E. R. QUEER (WRITTEN): The authors have presented a clear and concise statement of the factors affecting heat transmission in air spaces. The information on air space conductance values when the space is oriented in various angular positions is timely and of much value. Generally, experiments of this type have been carried out on small pieces of apparatus not having sufficient height to produce usable results. The authors by using a large sized experimental air space add much to the validity of their results.

First it might be pointed out that other investigators have evaluated separately radiation and convection in air spaces. Heilman in a paper on Surface Heat Transmission [3] describes a radiometer which he used to separate these values on external surfaces. In another paper [2] presented before the Society, radiation and convection have been separated by much the same method as used by the authors.

It is well to stress the fact that the ability of a surface or material to emit, absorb, or transmit radiant heat depends on the wave length of the radiation involved. In experiments [4] at Penn State it was found that when aluminum and white surfaces

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[3] Surface Heat Transmission. R. H. Heilman. *A.S.M.E. Transactions*—Fuels and Steam Power Division. Vol. 51, No. 22, p. 299.

[2] Importance of Radiation in Heat Transfer Through Air Spaces, E. R. Queer, A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 77.

[4] Surface Absorption of Heat from Solar Radiation. F. G. Hechler and E. R. Queer. *Refrigerating Engineering*. Feb. 1933.

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are exposed to the sun under similar conditions, the aluminum absorbed more heat. Likewise, glass which will readily transmit short wave infra-red rays or sun radiation, will intercept radiations at room temperature.

The authors have again confirmed the fact that heat transfer by convection across air spaces less than two feet high is greater per unit area than for higher spaces. Other investigators including the work [5] here at Penn State have found this to be true. On account of this fact it is more desirable to employ a high air space such as that used by the authors, in studying air space heat transmission coefficients. Small pieces of apparatus, such as have been used in the past undoubtedly introduce errors.

In Fig. 4 of the paper under consideration is shown a photograph of the apparatus used to study the variation of convection with height. No description is given of this equipment but it seems to be essentially the same as that used by Griffiths and Davis [6] in their investigation.

In Table 4 the authors give the equivalent thickness of insulating material for one bright aluminum surface in a vertical air space as 0.70 in. The value obtained at Penn State [7] indicated an equivalent thickness of about 0.50 in. for the foil used. In the latter case the foil was suspended in a 54 in. height air space so as to divide the original space into two 1.81 in. thick spaces. The higher insulation value obtained by the authors may be accounted for either by the very low emissivity foil used or by the fact that the aluminum foil on paper may not have been securely cemented to the steel plate. In the latter case a small air space between the paper and plate would introduce a certain resistance.

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[5] Loc. Cit. See Note 3.

[6] The Transmission of Heat by Radiation and Convection. E. C. Griffiths and A. H. Davis. The Engineering Committee of the Food Investigation Board. Special Report No. 9. 1922. His Majesty's Stationery Office, London.

[7] Discussion of "Insulating Value of Bright Metallic Surfaces"—F. B. Rowley, by E. R. Queer. A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 422.

No. 1066

PERFORMANCE OF FIN-TUBE UNITS FOR AIR COOLING AND DEHUMIDIFYING

By G. L. TUVE* AND C. A. MCKEEMAN** (*MEMBERS*), CLEVELAND, OHIO

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Case School of Applied Science.

THIS paper undertakes to summarize and to compare several of the methods which are at present used for expressing the cooling and dehumidifying capacities of surface coils. Methods are suggested for making rating tests of any coil, or for selecting a coil for a given service when such ratings have been made, and the accuracy of these methods is discussed.

Test results indicate that many of the simple methods of calculating surface-coil ratings are only approximations. It is also demonstrated however that accurate ratings call for much experimental work of a very precise nature, and for involved calculations to interpret the results. It is because this accuracy requires more time than the practicing engineer or the manufacturer can devote to the problem, that a program of cooperative research along these lines has been undertaken.

COMPLEXITY OF PROBLEM

Attention has been called in the previous reports on this research [1] to the large number of variables affecting the performance of a fin-tube coil. A list of these variables for an air-cooling and dehumidifying coil is given in Table 1. All of the variables indicated in this table have been investigated, to a greater or less degree, in this research project to date. In order to simplify the problem, however, the data and methods given in this paper have been applied only to copper fin-coils with cross-flow or mixed flow. (No true counterflow or true parallel-flow coils have been tested, and few such coils exist in practice).

Many scientific papers are available on the various phases of the subject of extended-surface heat transfer, and a number of the laws, by which the variables of Table 1 are related, have been developed. It is recognized, for instance, that the heat transfer in an air-cooling coil may conveniently be divided into three steps: (1) Heat flow from air to outside metal surface; (2) Heat flow through the metal; (3) Heat flow from the inside metal surface to the

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[1] For references, see Bibliography.
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1937, by G. L. Tuve.

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refrigerant. While these three steps may be related by writing the usual equation for the over-all coefficient of heat transfer in terms of the individual coefficients, this relation has not been extensively used in practical work, because the individual coefficients are not well known and are very difficult to determine.

Goodman [2] has recently outlined a promising method for finding both the air-side and the refrigerant-side coefficients. Pownall [3] and others have given a few data on the refrigerant-side coefficients, and Wells [4] has outlined a method for finding the *effective air-side coefficient*. These investigators have, in turn, built upon the earlier work of Knaus, Scanlan, Schmidt and

TABLE 1—VARIABLES AFFECTING FIN-COIL PERFORMANCE

	COIL
Surface-ratio:	Ratio of air-side surface area to refrigerant-side surface area.
Type of fins:	Round, square or continuous.
Shape of fins:	Plain or crimped, ribbon or wedge-shape.
Fin bond:	Integral, dipped, expanded, crimped, etc.
Material:	Copper, aluminum, steel, cast-iron, brass.
Depth and piping:	Depth of fins, number of tube-rows, tube spacing, counterflow, parallel flow, cross flow, mixed flow, etc.
	AIR
Velocity:	Mass velocity through free area, or linear velocity through face area.
Temperatures:	Dry-bulb, wet-bulb and dew-point.
Turbulence:	Reynolds number, local eddies, stratification and distribution.
	REFRIGERANT
Type:	Liquid or direct-expansion vapor.
Velocity:	Velocity of liquid, gas or vapor in tube.
Turbulence:	Produced by velocity or by mechanical means, such as spirals or corrugated walls.

others.[5 to 13] As a final result a number of conclusions have become fairly well established and are supported both by analysis and by experimental evidence. Some of these are given in Table 2. Certain of these statements are approximations, but they are much closer approximations than most of the assumptions given in Table 3.

PRACTICAL SOLUTIONS

Three methods are available for expressing quantitatively the performance of engineering equipment: (1) Performance tables; (2) Performance charts or graphs; (3) Performance equations.

But, as the number of variables increases above two or three, any of these methods becomes complex or cumbersome. Hence it has been the practice of fin-coil users to resort to various approximations which simplify the presen-

[2 to 13] For references see Bibliography.

TABLE 2—GENERAL STATEMENTS RELATIVE TO FIN-COIL PERFORMANCE

ITEM NO.	STATEMENT	REFERENCES		
		List No.[a]	Author	Page
1	Over a limited velocity range, the overall heat-transfer coefficient varies with a fractional power of the air velocity, i.e., $U = C\ V^n$.	10 2 1–*a* 1–*b*	McAdams Goodman Tuve-McKeeman Tuve	266 240–41 427 589
2	The heat-load ratio may be determined from the slope of a straight line on the psychrometric chart which connects the points representing the inlet and the outlet conditions.	9 3 8 4	Knaus Pownall Goodman Wells	23–82 211 90 665
3	The relations between dry-coil capacity and dehumidifying capacity may be expressed in terms of the characteristics of this straight line drawn on the psychrometric chart through the entering and the leaving air conditions.	6 4 9	King-Knaus Wells Knaus (See also Fig. 6 herewith)	283 665 25–82
4	With all other conditions constant, the total load varies with the wet-bulb temperature of the entering air, irrespective of dry-bulb or dew-point temperatures.	13 4	Scanlan Wells	197 665
5	The effective surface temperature corresponds to that dew-point temperature of the entering air at which condensation appears on or disappears from the air-side surface.	4	Wells	667
6	The difference between the effective surface temperature and the refrigerant temperature varies directly as the total load, irrespective of the nature of the load.	3 4	Pownall Wells (See also Fig. 3 herewith)	211 667
7	The heat transfer coefficient is increased by increasing air turbulence or local eddies in the approaching air stream.	1–*a* 6	Tuve-McKeeman King-Knaus	427 283
8	Separate tables may be prepared and used for the sensible-heat and the latent-heat transfer capacities of a coil.	1–*b*	Tuve (See also Fig. 7 herewith)	594

[a] See Bibliography.

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TABLE 3—COMMON APPROXIMATIONS RELATIVE TO FIN-COIL PERFORMANCE

ITEM NO.	ASSUMPTION[a]	FOR CHECKING[b] ACCURACY SEE:
1	A change in the number of rows of tubes (coil depth) does not change the heat-transfer coefficient.	Ref. 1*b*, p. 592
2	A simple factor may be used to correct the heat-transfer coefficient for the rate-of-flow of the refrigerant.	Ref. 1*b*, p. 593
3	The coefficient of sensible heat transfer is unaffected by the magnitude (or absence) of accompanying latent heat transfer.	Fig. 6
4	A simple correction factor may be used for wet-coil operation, dependent on temperature difference between entering dew-point and refrigerant.	
5	Air always leaves a fin-coil in the saturated state	Fig. 8
6	Air always leaves a fin-coil at a fixed relative humidity, irrespective of operating conditions.	Fig. 8
7	When a straight line is drawn on the psychrometric chart from entrance to exit air conditions, it will intersect the saturation curve at the effective surface temperature.	Ref. 4, p. 665 Ref. 5, p. 299 Fig. 3
8	A straight line on the psychrometric chart from entrance to exit air conditions represents also the succession of state-points, i.e., the condition curve for the air.	Ref. 2, pp. 228 and 250
9	The difference between effective surface temperature and refrigerant temperature is constant.	Ref. 4, p. 668 Fig. 3
10	The effective surface temperature is equal to the mean between the wet-bulb temperature of the exit air and the temperature of entering refrigerant.	

[a] Some of the assumptions are incomplete statements, but it is assumed that the reader will recognize the omissions.
[b] For references, see Bibliography.

tation of coil data. In Table 3 are listed a number of these common approximations.

In the following sections the results of a number of tests on cooling coils will be presented, and these will be correlated by showing how the test results agree or disagree with the statements given in Tables 2 and 3. Before presenting these correlations, however, it will be necessary to indicate the specific problems which the engineer is most frequently required to solve.

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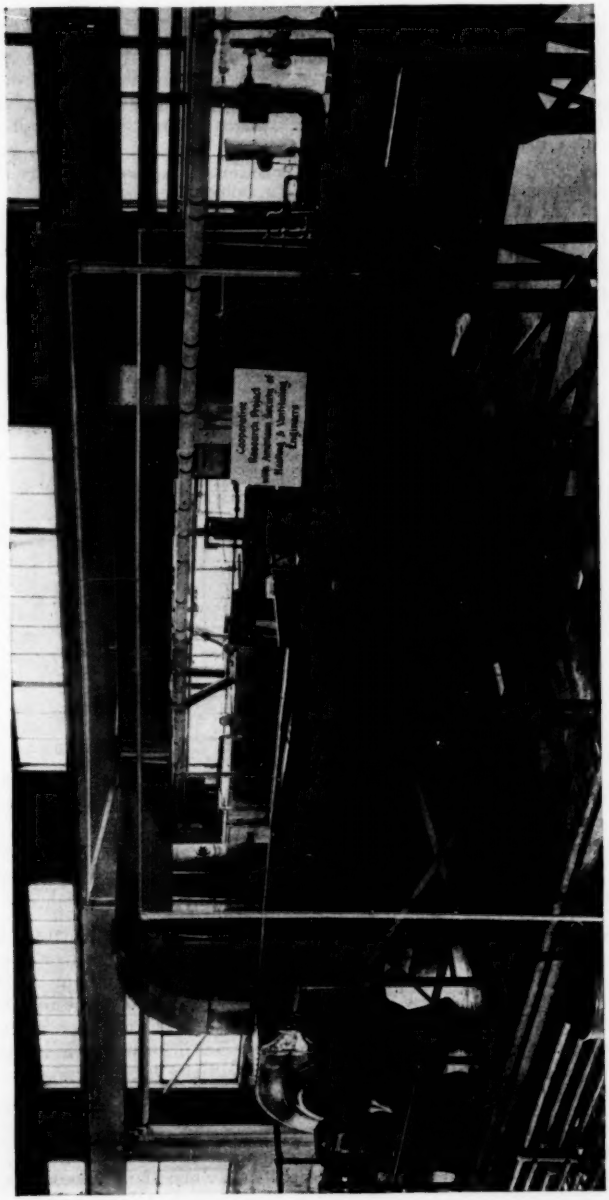

FIG. 1. GENERAL VIEW OF FIN-COIL TEST UNIT

COIL RATINGS AND COIL SELECTIONS

Three typical engineering problems will be considered:

(1) How should a coil be rated (i.e., its performance determined by a standard method), and how may the performance be expressed in order that one coil may be compared with another?

(2) If the coil performance under certain conditions is known, how may the performance under another set of conditions be determined?

(3) Given the outdoor and indoor air conditions, and the calculated sensible and latent heat loads, how may a coil be selected which will handle these loads and produce the desired conditions?

There are at present almost as many methods for rating a coil as there are coil types or designs. The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS might well look forward to the establishing of a Standard Rating

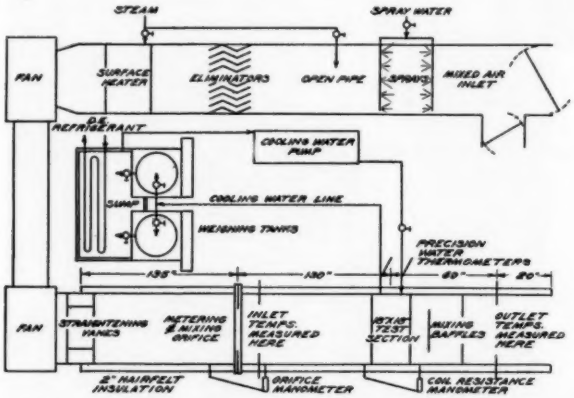

FIG. 2. LINE DIAGRAM OF APPARATUS SHOWN IN FIG. 1

Code, similar to those it has adopted for radiators, convectors and boilers. It is not too early for members to express their views on this subject, because much discussion will be needed in order to arrive at the best methods.

The problem of determining the changes in coil performance when the conditions are changed is only partially solved at present. If the change is merely one of air condition, refrigerant temperature, or air or refrigerant velocity, a close approximation to the new performance can be obtained. But for changes in coil design, in coil depth, in type of refrigerant, or in the type of flow as when changing from cross-flow to counter-flow, more information is needed and attention is called to the Program of Research outlined hereinafter.

The solution to the problem of selecting a coil for a given set of conditions will depend on the nature and the completeness of the rating-data available on the coil-type being considered. Many selections require compromises of one kind or another, and it must not be expected that when the air conditions and

the loads are fixed, the refrigerant temperature, air velocity, space requirements and similar factors may also be arbitrarily prescribed. This amounts to *telling the coil what to do*, rather than making a reasonable selection and then calculating what that coil *will* do. It is a simple matter to write specifications which cannot be met by surface coils at all. Special arrangements such as by-passing, reheating, etc., will enable surface coils to meet certain of these difficult requirements.

RESULTS OF TESTS ON TYPICAL FIN-COILS

The tests reported in the earlier papers on this project[1] were made with a draw-through arrangement taking air from the room, but the equipment has

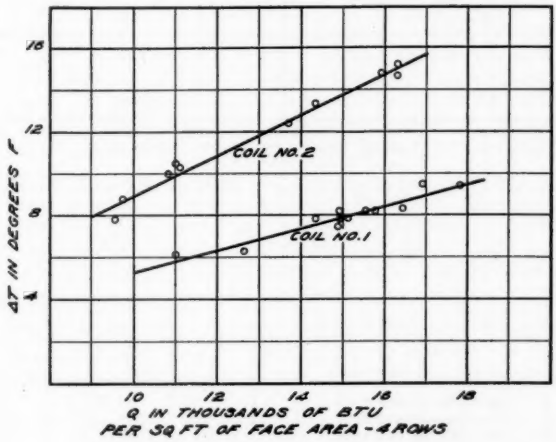

FIG. 3. TEMPERATURE GRADIENT ΔT BETWEEN REFRIGERANT AND AIR-SIDE SURFACE

now been rearranged for blow-through operation, with air supplied from a conditioner. Fig. 1 is a general view of the equipment, with the testing tunnel in the foreground and the air conditioner in the background. The diagram of Fig. 2 shows the relative arrangement of parts, with principal dimensions.

Results are given herewith on two of the four types or makes of coils which have been tested to date in this research. Results on the other two coils lie in the intermediate region, and will be presented in a later report. *Coil No. 1* was a 4-row thick-fin, round type, with a low ratio of fin surface to tube surface, and with a high-turbulence water path. *Coil No. 2* was a 4-row continuous-fin coil with thin fins, a large ratio of extended surface to tube surface,

[1] For references, see Bibliography.

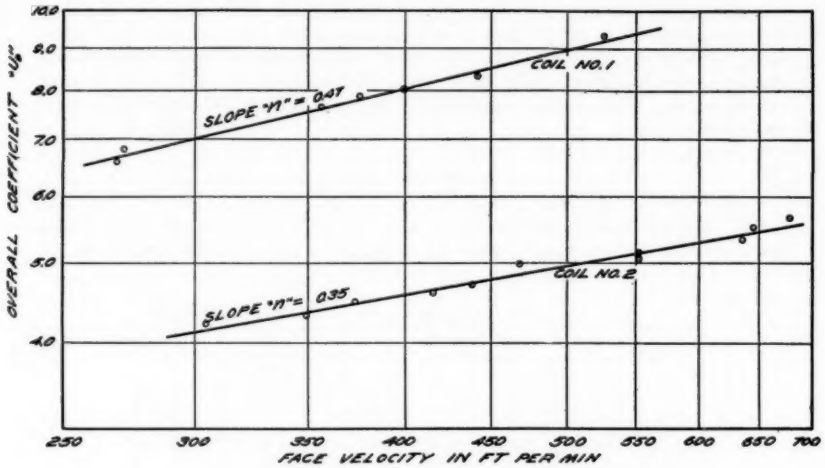

FIG. 4. OVER-ALL COEFFICIENTS OF DRY COILS PER SQUARE FOOT OF AIR-SIDE SURFACE

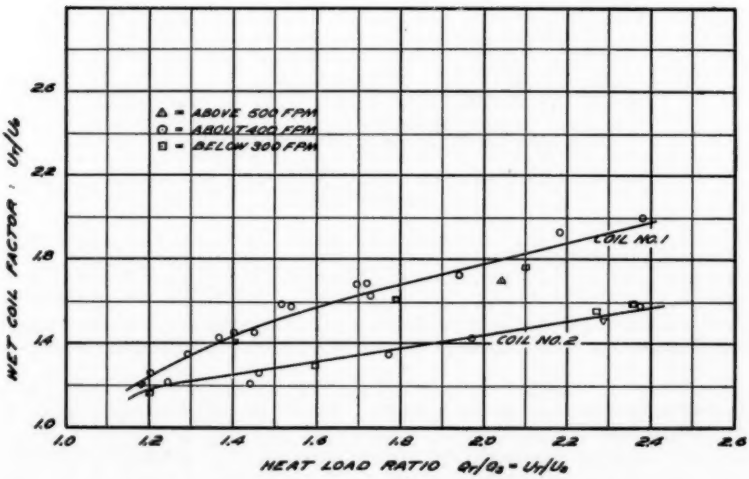

FIG. 5. TYPICAL WET-COIL FACTORS FOR TOTAL HEAT

and a smooth-tube water path. Most commercial coils with 4 rows of tubes will therefore lie within the region covered by these two coils.

Fig. 3 shows the temperature difference (or temperature gradient), between the refrigrant temperature and the effective surface temperature as defined in Table 3, item 7, and in Table 2, item 6.

Fig. 4 shows the dry-coil performance of these units over a range of face velocities from about 300 fpm to 700 fpm, and with water velocities in the tubes about 1 ft per second. The over-all coefficient is the heat transfer in Btu per square foot of surface area, per degree log mean temperature difference between air and refrigerant.

Fig. 5 gives the factor by which the dry-coil coefficient from Fig. 4 must be multiplied in order to obtain the total heat transfer coefficient with dehumidification. This factor is plotted against the load-ratio, total-heat load to sensible-heat load.

Fig. 6 shows several other methods of plotting the same data as that shown by Fig. 5. Some engineers wish to express the load-ratio in terms of sensible to total, or of sensible to latent, and the corresponding wet-coil factors are given in Fig. 6. If an engineer prefers to select coils on the basis of sensible heat only (instead of total heat transfer), the factors in the lower row of graphs should be used. All the curves of Fig. 6 give identical results, and they merely represent different methods of plotting the data of Fig. 5.

Fig. 7 gives the heat-transfer coefficients for latent heat only. Since the heat potential for latent heat transfer is the difference between dew-point (i.e., condensing temperature), and refrigerant temperature, the mean temperature difference for the latent-heat coefficient must be calculated on the basis of the dew-point temperatures.

Fig. 8 shows the difference between dry-bulb and wet-bulb temperatures at exit, and the approximate relative humidity of the exit air, for 4-row coils of usual design, based on the same two coils, and covering the range of heat-load ratios.

DISCUSSION OF DATA IN FIGS. 3 TO 8

In order to indicate the care with which the data here presented have been obtained, a number of the precautions in test procedure will be enumerated:

1. Water temperatures were read to 0.01 F, on accurately matched thermometers.

2. Outlet water was mixed by three 90 deg elbows before reaching the thermometers.

3. Air temperatures were read to 0.1 F on four equally-spaced thermometers at entrance, and four at exit.

4. Exit air was mixed by two orifices in series, with a target between.

5. Wet-bulb thermometers were continuously supplied with water through long precooling wicks, with capillarity assisted by gravity.

6. All temperatures were controlled within 0.3 F during each 60-min run.

7. Air quantities by heat balance must check air measurements by orifice within 5 per cent.

8. Dehumidification by heat balance must check dehumidification by weight of drip-water within 5 per cent.

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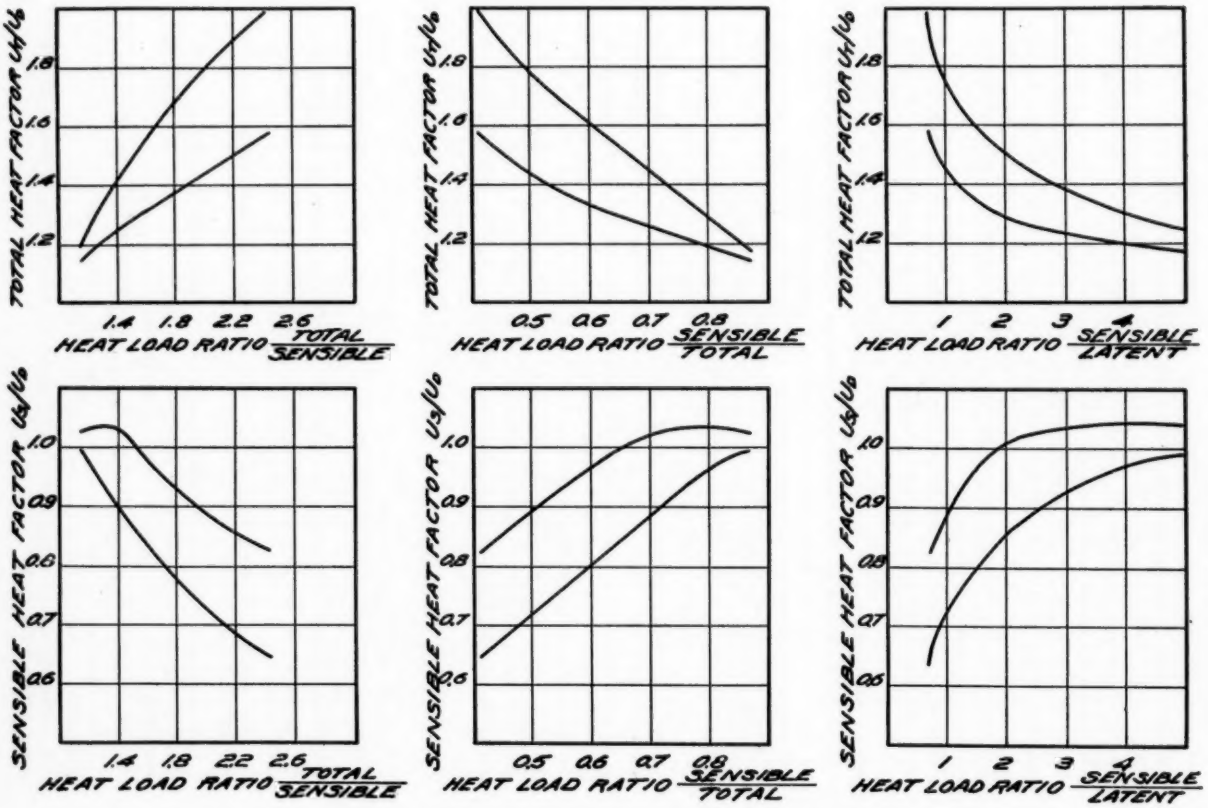

FIG. 6. SIX METHODS OF PLOTTING THE WET-COIL FACTORS

9. A complete set of readings was taken every 4 min for each 60-min run.

10. A preliminary equalizing period of at least one hour preceded each run.

On Figs. 5, 7, and 8 the approximate air velocities for the various tests are indicated as follows: squares indicate velocities below 300 fpm; circles indicate velocities around 400 fpm; triangles indicate velocities above 500 fpm.

It was found to be entirely possible to make repeat or duplicate runs on dehumidification, with all conditions steady and all check-readings satisfactory, and yet have the points separate as much as is indicated by Fig. 5. There is no doubt about an actual difference of performance between such runs. A similar difficulty has been reported by investigators in all fields dealing with surface condensation. Great differences have been shown to exist in the rate of heat transfer in condensation, depending on whether the condensation is being formed as *film condensation* or as *drop condensation.* It is the opinion of the authors that similar differences exist in the case of dehumidifying coils, depending perhaps on the condition of the surfaces, and their immediate-past history. Differences in the nature or completeness of condensate drainage from the fins also affect the performance. It is, therefore, not to be expected that ratings of dehumidifying coils can be set within the same tolerance limits as those used for dry-cooling coils.

APPLICATION TO PRACTICAL WORK

From data such as those plotted in Figs. 3 to 8, various methods of procedure are available for obtaining answers to the three questions already listed under Coil Ratings and Selections.

As a typical problem, consider the case of a fixed condition of the air entering the coil, and specified sensible and latent heat loads to be carried. This fixes a starting point on the psychrometric chart, and fixes the slope or direction of the line on this chart which connects the air conditions at entrance and exit.

It is then necessary to select the type of coil to be used, and the kind and method of application of the refrigerant. Assume, for example, that a style of unit such as the No. 1 coil in Figs. 3 to 8 is to be used, with water as the refrigerant, and a small water-temperature rise.

The remaining variables are: (1) Coil depth; (2) Refrigerant temperature; (3) Air velocity; (4) Face area; (5) Exit air conditions. In the present example, assume that a 4-row coil is to be used (since the graphs herewith are based on 4-row coils). The problem may then be solved either on the basis of a fixed mean refrigerant temperature, or on the basis of a fixed air velocity. In either case the exit air conditions and the face area will depend on the properties of the coil, and hence these conditions cannot be arbitrarily fixed.

Summarizing, the conditions for this example are:

Given: Total load; load ratio; entering air conditions; air velocity (or refrigerant temperature).

Required: Exit air conditions; refrigerant temperature (or air velocity); coil face area.

For a specific solution a face velocity of 450 fpm will be assumed, with a

FIG. 6. SIX METHODS OF PLOTTING THE WET-COIL FACTORS

total load of 60,000 Btu per hour, of which two-thirds is sensible heat and one-third latent heat, and air entering at 85 deg dry-bulb, 50 per cent relative humidity. Procedure by the various methods is as given:

Step 1. Locate the initial condition point on a non-logarithmic psychrometric chart. Draw a straight line through this point for a total-to-sensible heat-ratio of 1.5, at the proper angle for the particular chart used. This line may be either drawn through a second point which has been calculated from the psychrometric tables, or the table of angles supplied with the particular chart may be used.

Step 2. Using Fig. 8, determine the exit air conditions for $Q_T/Q_S = 1.5$. These are: Dry-bulb wet-bulb difference = 3.5 deg, relative humidity approximately 82.5 per cent. Referring to the psychrometric chart, the exit conditions for this coil are found to be: Dry-bulb = 64.7 deg, wet-bulb = 61.2. From the wet-bulb temperatures at entrance and at exit, the heat extracted from each pound of dry air is found to be 7.6 Btu.

Step 3. The load per square foot of face area may now be calculated by the heat-balance equation:

$$Q = \frac{\text{face velocity feet per minute} \times 60 \times 0.075}{\text{pound mixture per pound dry air}} \times \text{heat extracted per pound}$$

= 15,220 Btu per hour per square foot of face.

Then: Face area $= \frac{60{,}000}{15{,}200} = 3.95$ sq ft.

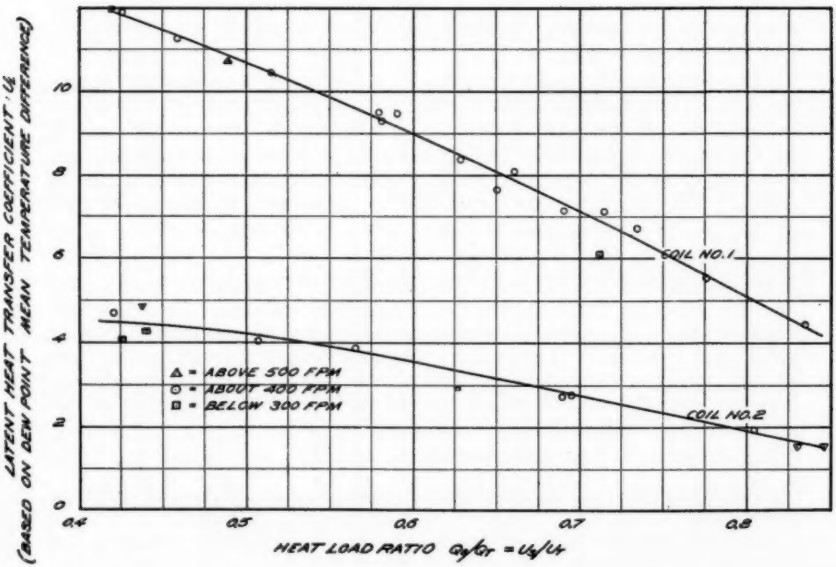

FIG. 7. LATENT-HEAT COEFFICIENTS BASED ON DEW-POINT TEMPERATURE DIFFERENCE

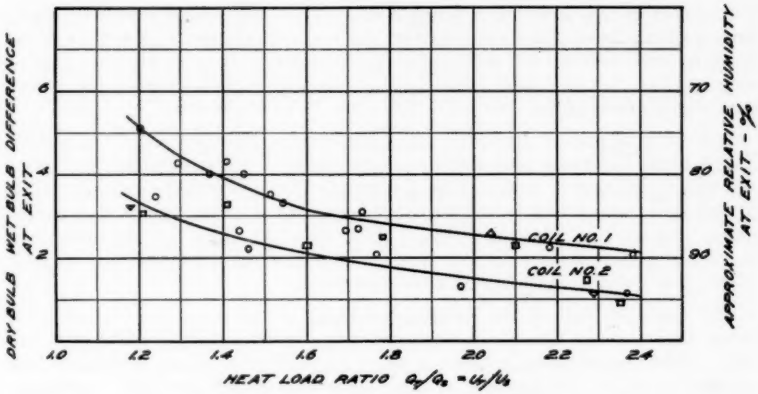

FIG. 8. AIR CONDITION AT COIL EXIT

Step 4. To determine the refrigerant temperature:

1. *Surface Temperature Method.* From Fig. 3, at this load, the difference between surface temperature and refrigerant temperature is 8.1 deg. The surface temperature (defined in Table 3, Item 7) from the psychrometric chart is 57.5 deg, hence the refrigerant temperature is 49.4 deg.

2. *Over-All Coefficient Method.* From Fig. 4 the dry-coil coefficient is 8.55. The wet-coil factor, total heat basis, from Fig. 5 is 1.51, hence the over-all coefficient for total heat, $U_T = 12.9$. Solving for the logarithmic mean temperature difference (dry-bulb basis), using 15,200 Btu per square foot of face and 51.7 sq ft of surface per square foot of face (as measured from this coil), the result is: $mtd = 23.75$ deg. Solving for refrigerant temperature (by formula, tables or chart), the result is 49.8 deg by this method.

3. *Latent Heat Method.* From Fig. 7, the latent heat coefficient $U_L = 7.75$ (based on dew-point temperatures). Solving for dew-point log mean temperature difference, the result is: $mtd = 12.6$ deg. The refrigerant temperature by this method is then found to be 49.5 deg.

While the example given uses the same known conditions to illustrate all three methods of solution, each method is best applied within a certain range. The selection of method depends upon which conditions are known and which are unknown.

CONCLUSIONS AND PROGRAM OF RESEARCH

The methods of coil selection illustrated by the examples are free from the cut-and-try procedure which characterizes several of the methods now in common use. The methods here presented also give definite solutions in accordance with the characteristics of the coil, instead of *telling the coil what to do.* It is a common fault to specify too many conditions and to expect a given type of coil to satisfy all of them. The above methods offer check solutions by which this common error may be detected.

It should be emphasized that all of these methods are subject to discrepancies arising from inaccurate construction on the psychrometric charts, and to disagreements between the various charts themselves. It is particularly easy to make an error of one-half degree or more in the surface temperature, where that method is used.

More data should of course be accumulated as rapidly as possible, to include a variety of coil types and of operating conditions. It is probable that after this has been done, a coil may be completely rated by merely making a few dry-cooling tests as shown in Fig. 4, because there is apparently a close correlation between Fig. 4 and the Figs. 3, 5, 6, 7 and 8. Data on this correlation are being prepared for a later report. These methods therefore have considerable promise as bases for Test Code procedure.

The main purpose of the research program in the immediate future is to refine the accuracy and to increase the comprehensiveness of the results obtained from the methods outlined in this paper. Data on other depths of coils are necessary, as well as data on other coil designs and operating conditions. Some additional results have already been obtained and are being completed as rapidly as possible.

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DISCUSSION

WILLIAM GOODMAN[14] (WRITTEN): Professors Tuve and McKeeman have formulated and clearly stated the factors which must be considered in rating and selecting coils. The clear formulation of a problem is a long step toward its satisfactory solution. The emphasis in this paper upon the subject of "telling a coil what to do" is especially worthy of mention. Many do not seem to realize that with any given initial velocity and refrigerant temperature, and any given initial condition of the air, the performance of a coil is fixed insofar as tonnage, and final dry- and wet-bulb temperatures are concerned.

No sharp distinction seems to be made between *selecting coils* and *rating coils.* They are two separate and distinct problems. By the rating of coils is meant the computation of coil performance, under various conditions, from experimental data. By selection of coils is meant the choosing of a coil for a particular application from the rating data previously prepared by the manufacturer. Regardless of the complexity or simplicity of the method used for computing the *rating* of coils, the method for *selecting* coils can be made so simple that no computations are needed whatsoever. It is not a difficult matter to present the rating data in the form of either curves or tables in such a simple manner that the selection of coils for various conditions can then be made easily and quickly. With such curves, or tables, it is also a simple matter to determine the performance of coils under conditions other than those for which the coil was originally selected. For this reason, it seems to me that the sole criterion on which the selection of a rating method should be based is the accuracy with which it can be used to predict actual results.

The method presented in this paper for selecting coils is simple. Moreover, the problem presented shows that this method does not involve any cut-and-try processes when selecting a coil. However, manufacturers are usually faced with the problem of computing the performance of a given coil. In other words, starting with a coil of a given number of rows, the initial air condition and velocity, and the refrigerant temperature, it is necessary to compute the leaving air condition and the refrigerating capacity of the coil. This cannot be done with the method presented in this paper without using a cut-and-try solution. Although such a cut-and-try method is not difficult, and can be used when only a few computations are to be made, it becomes impractical when the performance of a large number of coils under widely varying conditions must be computed, which is often the case in the computation of tables and curves for catalogs.

HARRY SCHMIDT (WRITTEN): In the test proceedings under paragraph 8 I note

[14] The Trane Co., La Crosse, Wis.

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the following: "Dehumidification by heat balance must check dehumidification by weight of drip water within five per cent."

I assume that all test points reported in these tests fall within this range, and that the coils tested were of such a nature that complete drainage occurred during the test.

However, due to the nature of the coils and other factors it is not always possible to check dehumidification by weight of drip water within the given limits. One, therefore, cannot expect that ratings of all dehumidifying coils can be held within the same tolerance as those used for dry cooling coils.

J. G. O'FLAHERTY[15] (WRITTEN): The authors of this paper are to be commended for the comprehensive manner in which they have presented a very difficult subject. Obviously a great deal of careful thought and effort have gone into the making of this report, and it is particularly gratifying to note the practical trend of their research for the problem is one which readily lends itself to a most complex and theoretical treatment, due to that ever present and complicated mechanism, the condensation of a vapor.

The engineer engaged in air conditioning application, is only concerned with securing equipment which will meet his requirements and perform in accordance with manufacturers' ratings. It therefore behooves the manufacturer in cooperation with the research engineers to place before consulting and designing engineers such data that will enable them to make their own coil selection with reasonable accuracy and simplicity. This paper in the writer's opinion is a definite step in that direction.

The presentation of the performance of fin-tube coils is obviously most desirable in tabular form, and it is equally obvious that to compile dehumidification performance tables for all conditions likely to be encountered would seem to present a monumental task, and could only be accomplished after very exhaustive tests. No method has as yet been presented which would permit the compilation of such tabular data from a few simple dry-cooling tests. However, the authors may be able to give us some information on this point.

One thing in particular is very clear and has been emphasized by the authors in this and preceding papers, and that is the apparent need of some uniform method of rating and selecting coils. The writer heartily endorses the authors' statement that a standard rating code should be established and as quickly as possible. If a standard rating code were etablished then it would follow that a standard selection method would be established.

Expressing the wet-coil factor as a function of the heat load ratio ($Q\text{-}T/Q\,S$), Fig. 5, seems to the writer preferable to expressing it as a function of temperature differentials. In the latter case the entering refrigerant temperature must be known or assumed, whereas the heat load ratio is known or can be determined without any assumptions. I would like to ask the authors if they have made any investigations as to the effect of horizontal tubes on the wet-coil factor since I assume all tests were made with tubes vertical. This would apparently affect the type of condensation, particularly on a circumferential fin-tube unit.

Fig. 8 also merits some attention, since it appears to contradict the prevalent idea of a fixed dry-bulb wet-bulb difference at exit. It also shows quite clearly that the relative humidity at exit is not fixed or constant, but that both these values are functions of the heat load ratio, which is only logical. Is there much variation in these values for different coil depth?

Referring to the typical problem given as an example of the application of the various data, it is to be noted first that the air volume is not stated. This of course

[15] Canada Vulcanizer and Equipment Co., Ltd., London, Canada.

is easily determined once the required face area is found, but might this condition not lead to some confusion? In the majority of cases the air volume may have previously been determined by the space requirements. Are we to infer that if the air volume is given the face or air velocity then becomes an unknown to be determined?

We also note that the authors use the term "mean refrigerant temperature." Are the values derived from the solution of this order? I would like to have some further explanation as to why they prefer the use of the mean value. It simplifies the determination of the *mtd*, but again it seems to be customary in present day practice to refer to the refrigerant temperature, *i.e.*, when water is the medium, as being "initial temperature." When such is the case, in order to determine the "mean" and we assume the arithmetic mean, we have to enter the realm of assumptions and approximations again, which is always to be avoided if possible. Consider the implied assumptions if only initial temperature is given: first water temperature rise must be assumed, which in turn, given total heat load, determines the water rate and hence the coefficient of transmission for this rate or water velocity, all based on an initial assumption. Is this procedure justified? The authors have given plenty of scope to the method of determination of unknown values, and I would like to know the approximate range for the applications of the various methods.

Have the authors made any investigation or determination as to the maximum differential between entering dew-point and refrigerant temperature beyond which no dehumidification is to be expected? Fig. 3 would seem to give some reason to expect that some limit of difference must exist, dependent on the effective surface temperature.

Since the authors agree that discrepancies might occur due to inaccurate construction on the psychrometric charts, would it not be possible to prepare a table of heat load ratios, based on entering air conditions and mean refrigerant temperatures from which exit conditions could be determined arithmetically? It very often happens that the engineer who is called upon to make the coil selection is given only the value of the total heat load, with no indication as to the ratio of sensible or latent heat. It would then become very difficult to make a selection using any of the data outlined, unless he assumed a value for the heat load ratio. If these ratios could be presented in tabular form, having been derived from test data peculiar to any specific coil, it would seem to me to meet a very evident need.

I am in agreement with the authors' statement that it is a common fault to specify or fix too many conditions which a coil is expected to satisfy. This brings to our attention once again the apparent need of some suitable basic code, which should not only govern test and selection procedure but also specification procedure. Since discussion of the adoption of such a code has been invited, perhaps it would not be amiss at this time to offer some opinions. There seems to be no doubt whatever among the profession in general, that something should be done towards the standardization of cooling data. But what? Surface cooling has made very rapid strides during the past few years, and no doubt will continue to gain favor, so why not accord this important item of air conditioning equipment the recognition and prestige which is its due?

Since the computation and compilation of cooling test data are very contentious matters, would it not be feasible and practical to establish a central testing bureau similar to the procedure adopted for convectors? Such a central bureau could be selected, which is already well equipped to carry on without any additional equipment costs. The manufacturers I am sure should welcome such an opportunity, as such a procedure would entirely eliminate all elements of prejudice and partiality. All published data, which have been compiled from such tests, would bear the seal of

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approval of the Society, and it is remarkable to note the increased prestige accorded a product which is rated in accordance with codes prescribed by this Society.

The matter of test procedure and method of selection will require considerable discussion, and it is to be hoped that some such discussion will be evoked at an early date. If I might make the suggestion, would it not be a good idea to have appointed a temporary committee to investigate the possibilities of the establishment of some form of basic code? This at least would be a start in the right direction.

A. B. NEWTON [16] (WRITTEN): I have read this paper with a great deal of interest, and wish to compliment the authors on their painstaking test work, and the presentation of an interesting and valuable paper.

The large variety of dehumidifying coils available for air conditioning work has created a very definite need for unified rating data for all of them which meet the needs of field application work. A unified method of rating seems to be particularly needed at present, and as the authors mention it is necessary to eliminate cut-and-try procedures and those methods which "tell the coil what to do." Rating methods which involve individual surface heat transfer coefficients are primarily of interest to the manufacturer of coil equipment in analyzing proposed coil designs, while in application of coils we are concerned with the over-all performance and field work is greatly facilitated by data based on over-all heat transfer. I believe the authors' use of the latter method is a definite step forward, as the number of variables faced in the field would not justify general use of the individual surface coefficients even if men in the field could be made to understand their use.

As to the superiority of any one method of presenting data, it should be borne in mind that when applying coils we are generally interested in the performance of an entire air conditioning system. Therefore selection of coils for direct expansion systems is made easier by those methods which permit quick determination of the required refrigerant temperature for the sensible and latent loads involved. In selecting coils for chilled water systems which have a substantially constant refrigerant temperature, a rapid determination of the required fin surface temperature and of the difference between fin surface temperature and average water temperature is desirable. Considering these points, it appears that of the three methods which the authors have analyzed, the method involving Fig. 3 is most desirable.

A code for rating coils at various operating conditions is sorely needed, as a wide variety of operating conditions is met in practice, many of which are not yet well defined. Along this line it may be well to note in passing the need for more specific information relative to variations in the sensible and latent loads experienced in different types of installations and in different parts of the country. Such information, in conjunction with easily used coil data, helps determine within what limits control of temperature and relative humidity is possible as the load conditions vary.

I believe one very important problem which the authors mention is usually neglected; namely, the problem of condensate drainage. Whenever the source of refrigeration to the coil is alternately supplied and turned off in the operating cycle, the length of the cycles has a major effect on the coil performance, and some method of estimating this effect is highly desirable. I have observed several recent jobs in which the moisture condensed during the "On" period did not run off the fins at all, but instead re-evaporated into the air stream during the "Off" period. Naturally the result is a serious lack of dehumidification, and it appears that some research work to determine the effect of the length of cycle on the dehumidifying capacity of coils tested is needed. Such variables as flat vs. corrugated fins, circular or square fins vs. continuous fins are undoubtedly having their effect in this action.

[16] Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.

With regard to the authors' statement that the required surface temperature is difficult to obtain on psychrometric charts within plus or minus one-half degree, it is perhaps worthy of note that non-logarithmic charts are now available on which the angle of intersection of the line representing the ratio of sensible and latent loads with the saturation line has been steepened by the method of constructing the chart. This somewhat increases the accuracy of the determination.

There is, perhaps, little preference among the various methods of expressing the ratio of sensible and latent loads. However, it usually happens that the greatest load conditions occur at the time of greatest sensible load, and therefore I believe there is some slight practical advantage in using the ratio sensible to latent heat. This ratio will usually be greatest when the load is greatest, and smallest when the load is smallest. Eventually it may be well to choose some standard term for the ratio which is accepted for general use.

R. H. NORRIS [17] (WRITTEN): Professors Tuve and McKeeman are to be commended for the program of research which they are carrying on to obtain better and more standardized rating methods for the performance of fin-tube coils. The present paper is particularly useful for its convenient summary of many of the facts and principles involved in this very complex problem.

In regard to a standard code of rating, the present Standard Method of Rating and Testing Air Conditioning Equipment by the Joint Committee on Rating Commercial Refrigerating Equipment (sponsored by the *A.S.R.E.*) is an initial step in the direction of standardization of rating for finned coils, but provides for comparison under only a few sets of conditions. The establishment of a more comprehensive Standard Rating Code by the A.S.H.V.E. would seem very desirable.

Much of the present paper is concerned with over-all performance, with water on the inside of the tubes. The recent paper by Wells (reference 4 in the present paper), on the other hand, is restricted to the case of an evaporating refrigerant inside the tubes. It would seem that one object of the proposed program of future research should be the separation of the air-side and refrigerant-side performance data in some manner whereby, from suitable curves, tables, or equations for the air-side performance, the over-all performance could be readily computed for either hot or cold water, evaporating refrigerant, or steam, on the inside of the tubes, provided data for the behavior of the latter were obtained. For this purpose, it is suggested that the research program include the following objects:

1. A correlation of the effective surface temperature as defined by Wells and used by the authors, with the actual tube temperatures, so that the data already available on the heat transfer coefficients of water flowing in tubes can be applied to predictions of the over-all performance when different water velocities or temperatures are used from those used in tests.

2. Determination of the allowable range of conditions and the expected accuracy when a single average value effective surface temperature is used instead of the analysis of the varying surface temperature proposed by Goodman (reference 2 in the present paper).

3. Determination of the tube-to-refrigerant heat transfer coefficients for evaporating refrigerants with different velocities and arrangements of flow so that these coefficients can be used with air-side performance data for predicting performance. Study of the effect of superheating, if any, of the refrigerant vapor, near the outlet, and of the effect of pressure drop of the refrigerant should presumably be included.

In regard to the concept of the effective surface temperature, comment on the wording of the definition may eliminate future confusion. In item 5 of Table 2 the authors give the definition: "The effective surface temperature corresponds to that dew-point temperature of the entering air at which condensate appears on or disappears from the air-side surface." Wells, on the other hand, shows that a plot of moisture content of the air versus moisture removal is approximately a straight line (for constant inlet wet-bulb temperature and air velocity) and he uses for the

[17] General Electric Laboratory, General Electric Co., Schenectady, N. Y.

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effective surface temperature the intercept of this straight line on the axis of zero moisture removal. This intercept would agree with the authors' definition only if the actual performance followed this straight line all the way to zero moisture removal. Wells comments, however, that at very low values of moisture removal, the linear relation is not accurately applicable because of the non-uniformity of the surface temperature over the coil. It would seem preferable, therefore, to use Wells' definition, rather than the authors', in order that the advantage of the straight line correlation be retained.

H. B. POWNALL[18] (WRITTEN): Professors Tuve and McKeeman certainly could not have picked a better time to present a summary of the various methods used for rating air cooling and dehumidifying surfaces. They hit the nail on the head in their statement that there are at present almost as many methods for presentation of ratings of a coil as there are coil types or designs. I think we will agree that the time is at hand for preparing some means for a standard method of presenting data pertaining to coil performance.

Their presentation of Tables 2 and 3 in this paper, tabulating statements which are theoretically true, or have been definitely proven by past tests in one table and by tabulating common approximations in another table, is an excellent start towards preparing data necessary for manufacturers to get together and decide upon one method of presenting data which could be approved by all.

In their example the authors show three distinct methods of attaining the same answer for their problem, which has proved something that has been in my mind for some time. That is, that regardless of the many methods of attacking the problem, in the majority of the cases we will all end up at the same place. In other words, when sufficient test data are taken and interpreted in curve form, it is impossible to go far wrong if, when using these curves, we can check back to test data. Of course, this assumes that the curves are not extended on either end beyond the test points. This means that we are all merely using different methods of interpolation of test data, and although one plot will look totally wrong to one accustomed to using a different method, we cannot criticize the author providing it brings him back to test points.

For example, I could take exception to Fig. 5, in which the authors have plotted ratio of total k to sensible k against ratio of total work to sensible work. In my paper referred to by these authors as reference 3, I showed a plot very similar to this in which I made the statement that this ratio held for any type surface from bare pipe to fin surfaces of very high ratios of secondary to primary surfaces. I also showed that as a ratio of total tons to sensible tons increased, the sensible had a tendency to fall away from the so-called bone-dry obtained upon a coil with no condensation. I made the statement that this curve had been obtained through tests, and in the paper I showed the reason for this phenomenon by referring to the psychrometric chart and proving that after a coil started condensing moisture from the air the dry-bulb mean temperature difference was no longer the driving force for the total heat and had to be corrected accordingly.

Thus, these authors and myself have both plotted certain results against the same coordinates and have obtained different curves. However, by using these curves in combination with other plots from the same test data we are both able to come back to our test data in the end. For example, Professor McKeeman's and Professor Tuve's Fig. 5 must be used in conjunction with their Fig. 8, whereas, my own curve had to be used with certain other data presented in my paper.

I have merely cited this example as further proof of the need of an accepted

[18] Mech. Engrg. Dept., York Ice Machinery Corp., York, Pa.

standard method of presenting data for the rating of air cooling surfaces. This standard method of presentation would relieve any doubt in different purchasers' minds when making comparisons of various type surfaces. It would eliminate any "telling the coil what to do." It would also tend to stop the writing of specifications by purchasers and their engineers which cannot be met by surface coils at all, which move I think we all agree is the most important.

C. F. WOOD: Professors G. L. Tuve and C. A. McKeeman are to be congratulated on the common-sense manner in which they have handled the confusing problems encountered in the determination of capacity of fin-tube units.

In reviewing the information presented in this paper, it appears that all tests have been run with constant water velocity, hence constant internal coefficient of heat transfer.

It has been the experience of the writer that trends established without consideration of the relative magnitude of the three basic coefficients of heat transfer may be misleading, also that to predict performance of an evaporator accurately, it is necessary to have reasonably accurate data on the three transfer values, namely: refrigerant transfer, metal transfer and air transfer.

In view of this experience, it is the recommendation of the writer that there be included in future work on this project a study that will lead to the determination of and isolation of these coefficients of heat transfer.

W. L. FLEISHER: In connection with the paper, by Professor Tuve and Professor McKeeman, I want to bring out a very simple method which I feel can be utilized to a satisfactory extent in the determination of the performance of fin-tube coils.

This method consists of taking dry-bulb and wet-bulb readings of entering air and dry-bulb and wet-bulb readings of the leaving air. The total heat between the entering and leaving wet-bulbs would then give the heat removed by the coils. From these readings, of course the dew-point conditions of the entering and leaving air could be determined and the amount of sensible and latent heat removal calculated. Then, by varying the entering conditions and repeating the tests or by changing the velocity over the finned surfaces, new relationships between latent and sensible heat removal could be determined in order to fix the true values of coils under varying conditions.

H. F. HUTZEL: I have read with much interest the very constructive analysis of cooling coil performance by Professors Tuve and McKeeman. The results of their investigation disclose the effect of the many variables which enter into coil performance.

In attempting to apply their method of coil selection it seems to me, however, that there are some limitations to the adaptation of their suggested method of coil selection. Whereas in the example given, the authors have been able to determine the face area of the coil and refrigerant temperature (in this case water), there is no means of determining the temperature rise of water nor the quantity of water to be circulated.

The authors refer to coefficients, shown in Fig. 4, as being limited to a water velocity in the tubes of about one foot per second. This limits the application of data presented in this paper to not only specific types of coils but also specific sizes. If these data were applied to identical types of coils, but longer in direction of tube length, then the results would be in error for the reason that in this case the water velocity would change and this would affect the internal tube coefficient.

PROF. G. L. TUVE: In answer to Mr. Goodman, the authors readily agree that the problem of coil rating is different from that of coil selection. Perhaps in this paper the viewpoint of the coil *user* has been emphasized, with too little attention

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to the viewpoint of the coil manufacturer. The two viewpoints are placed in sharp contrast by the comments and discussions which have been submitted.

Mr. Schmidt raises an important point when he states that the ratings of dehumidifying coils cannot be held within the same tolerance as that applying to dry-cooling coils. The experimental difficulties in a dehumidifying test are great, and accurate field tests are almost impossible.

Answering Mr. O'Flaherty, the authors cannot as yet prescribe a coil-rating method requiring dry-cooling tests only, but are inclined to think that further research will produce such a method. Since the tests reported in this paper were all made on 4-row coils with tubes horizontal, the authors cannot make any specific statements regarding the increase in relative humidity of the exit air when using a deeper coil. These tests were purposely limited to conditions involving a small temperature rise of the water refrigerant. For counterflow coils, with large differences between inlet and exit temperatures, more data are needed and better methods can probably be developed.

While the authors concur with Messrs. Norris, Wood, Hutzel and others regarding the need for accurate surface or film coefficients, they are also inclined to agree with Mr. Newton that the over-all coefficient will continue to be used by application engineers because of their familiarity with this type of analysis.

It is pleasing to see the hearty agreement with the suggestion that we need to standardize our methods of coil rating, testing and selection. Perhaps Mr. Fleisher makes it sound too simple, but the suggestions by Mr. Norris and Mr. Pownall have added a few precautions.

No. 1067

THE COOLING AND HEATING RATES OF A ROOM WITH DIFFERENT TYPES OF STEAM RADIATORS AND CONVECTORS

By A. P. KRATZ,* M. K. FAHNESTOCK,** AND E. L. BRODERICK † (*MEMBERS*)
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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the University of Illinois.

THE objects of the investigation were: (1) to determine the influence of different types of direct steam radiators and convectors on the rate of cooling of a room under winter service conditions after the steam supply to the heating units was turned off; (2) to determine the influence of different types of direct steam radiators and convectors on the rate of heating the same room from a stated minimum temperature after the steam supply to the heating units was turned on; (3) to correlate the results with the physical characteristics of the heating units.

The investigation was made with six units, consisting of two direct cast-iron radiators, two convectors with cast-iron heating units, and two convectors with non-ferrous heating units. Although it should be recognized that the results of tests of this type, made under unsteady conditions, are influenced by the thermal capacity of the room and the contents of the room in which the tests are made, it is probable that the relative performances of the different units are independent of the test room, and that comparisons based on these relative performances are therefore valid. The actual magnitudes represented by the test results apply only to the room in which the tests were made and, with the exception of those obtained under equilibrium conditions, would require modification before being applied to other rooms and under different conditions.

DESCRIPTION OF APPARATUS

The room heating testing plant, in which the tests were conducted, has been

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described in detail in previous publications.[1, 2, 3, 4] A plan and an elevation section of the plant are shown in Figs. 1 and 2. In brief, the plant consists of a large cold room with the walls, floor and ceiling insulated with two layers of 3-in. corkboard; two test rooms 9 ft x 11 ft with 9 ft ceilings; refrigeration equipment for maintaining sub-zero temperatures in the cold room; a complete thermocouple system for observing temperatures; steam, hot-water and electric service for testing units with different heating mediums; an eupatheoscope for determining the equivalent temperature; and controllers and recorders for observing data over periods of continuous operation.

Since the test rooms are located in the corners of the cold room, as shown in Fig. 1, two walls of each test room are formed by the insulated walls of

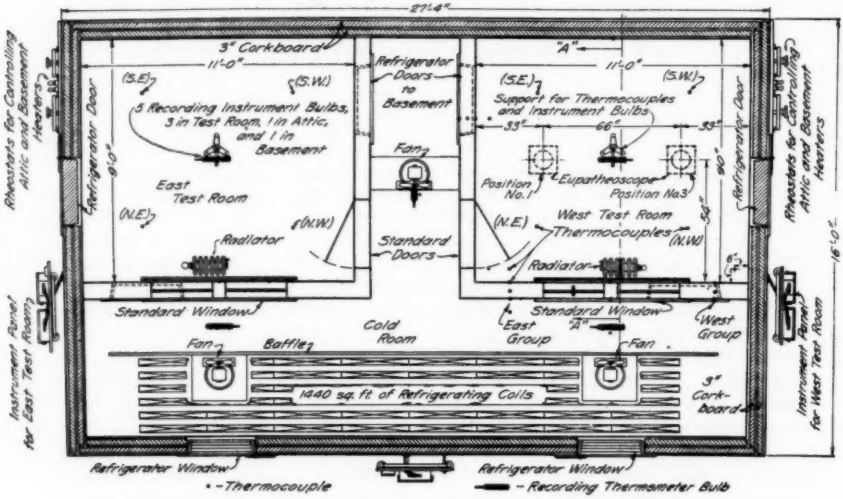

FIG. 1. PLAN SECTION OF LOW TEMPERATURE TESTING PLANT

the cold room. The other two walls of each test room are exposed to the refrigerated space, thus simulating a corner room of a building with two exposed walls. These walls are of standard frame construction, consisting of $\frac{5}{8}$-in. redwood siding, building paper, $\frac{3}{4}$-in. thick tongue and groove yellow pine sheathing, 2 x 4 yellow pine studding, and $\frac{3}{8}$-in. wood lath with $\frac{1}{2}$-in. gypsum plaster. As shown in Fig. 2, each room is provided with an attic and basement space equipped with electric heaters for the purpose of regulating the air temperatures above the ceilings and below the floors. The ceil-

[1] Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, S. Konzo, and M. K. Fahnestock, A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 79.

[2] Steam Condensation an Inverse Index of Heating Effect, by A. P. Kratz and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 475).

[3] Performance of Convector Heaters, by A. P. Kratz and M. K. Fahnestock, A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 351.

[4] University of Illinois Engineering Experiment Station Bulletins Nos. 192 and 223, and Reprint No. 1.

ings are made of ⅜-in. wood lath and ⅜-in. gypsum plaster with no flooring in the attic. The floors are of standard 2¼-in. x 13/16-in. yellow pine flooring over building paper placed on ¾-in. thick tongue and groove sub-floors. Each test room has one double window 4 ft-6 in. by 5 ft overall and one door 3 ft by 7 ft overall placed in the exposed walls.

The location of the direct expansion refrigerating coils in the cold room,

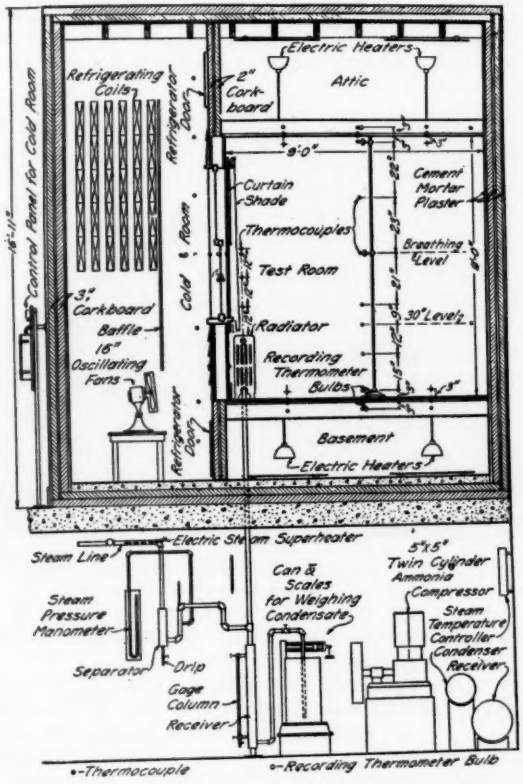

FIG. 2. ELEVATION SECTION OF LOW TEMPERATURE TESTING PLANT

and the ammonia compressor and condenser on the lower floor of the Mechanical Engineering Laboratory are also shown in Figs. 1 and 2. The refrigerating apparatus has sufficient capacity to maintain sub-zero temperatures in the cold room when the temperatures in the test rooms are approximately 70 F. Oscillating fans located beneath the refrigerating coils are used to direct cold air against the north wall of each test room at a velocity approximately equivalent to that of a 10 mph wind.

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As all of the tests reported in this paper were conducted in the west test room of the room heating testing plant, hereafter, all references to a test room, unless otherwise stated, shall be understood to apply to the west test room. Fig. 2 shows the locations of some of the thermocouples available for observing air temperatures at various levels in the center of the test room and in the cold room, and surface temperatures on the walls, floor, and ceiling. The leads from thermocouples are formed into cables and connected to a common switchboard located on the lower floor of the laboratory directly beneath the cold room. The use of thermocouples permits the observer to note temperatures without entering the test room and thus disturbing the conditions, and they

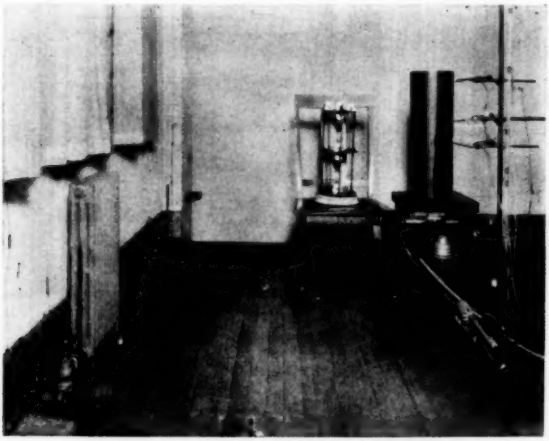

FIG. 3. THE EUPATHEOSCOPE WITH HEATING ELEMENT REMOVED

also make it possible to obtain surface and air temperatures in locations not accessible to thermometers.

The one-pipe steam connection for supplying steam to the units, and the condensate receiver, tank, and scales for determining the heat output of the units are shown in Fig. 2. Each unit was vented through a ⅛-in. pipe leading from the unit to the lower floor of the laboratory, and a constant rate of venting was maintained by means of an orifice 0.059 in. in diameter located in a pipe-cap on the discharge end of the vent pipe. Care was taken to make the vent connection to the units in such a manner that complete venting occurred without the loss of any unweighed condensate through the vent line.

The eupatheoscope, shown in Fig. 3, is an instrument used for evaluating the combined effect of radiation and convection in a heated space in which the air and the surfaces of the surroundings, such as radiators, furniture, walls, windows, etc., may be at different temperatures, in terms of an equivalent temperature consisting of the dry-bulb temperature of still air in a space in which the air and surroundings are at the same temperature. A complete description of this instrument, including a discussion of the rate of heat loss from

the body and its correlation with human comfort, has been given in a previous publication.[5] Briefly, it consists of a sizeable body consisting of a blackened hollow copper cylinder mounted vertically on a stool, and inside of which is an electric heating element for maintaining the surface temperature of the cylinder. Thirteen thermocouples made of No. 34 B and S gage copper and constantan wire are embedded at regular intervals in the surface of the cylinder for observing the average temperature, which is maintained constant at 83.0 F by adjusting the electric power input to the heating element. From the power input, measured with a sensitive wattmeter, the heat loss from the copper cylinder may be determined in Btu per square foot per hour. By referring to the original calibration curve which was established by determining the unit heat loss from the cylinder when placed successively in a number of uniform environments having different dry-bulb temperatures, the combined effect of radiation and convection in the non-uniform environment can be expressed in terms of an equivalent temperature. This equivalent temperature consists of the dry-bulb temperature of a uniform environment which would give the same unit heat loss from the cylinder. For all of the tests, the eupatheoscope was located in position No. 3 as shown in Fig. 1.

The following six units used in the tests were regarded as being representative of characteristic types of commercial units: Unit No. 1, an 8-section, 26-in., 5-tube, direct cast-iron radiator with 2½ in. between the centers of the sections; Unit No. 2, an 18-section, 25-in., 3-tube, direct cast-iron radiator with 1½ in. between the centers of the sections; Unit No. 3, a convector with a cast-iron heating element 6 in. high; Unit No. 4, a convector with a cast-iron heating element 14⅜ in. high; Unit No. 5, a convector with a non-ferrous heating element 5¼ in. high; and Unit No. 6, a convector with a non-ferrous heating element 2 in. high. Additional information concerning the physical dimensions of each of these units is given in the insets in Fig. 4 and in Table 1.

In previous tests made in the room heating testing plant a temperature of 68 F at the 30-in. level in the test room was maintained under equilibrium conditions. However, for the purpose of these tests, in which the steam supply to the units was to be turned off until the temperature in the room dropped to 60 F, it was recognized that in order to bring this air temperature back to 68 F in a reasonable length of time, units with capacities slightly larger than those required for steady operation would be necessary. The units used were, therefore, selected with sufficient capacity to maintain as nearly as possible 71 F at the 30-in. level under equilibrium conditions.

Test Procedure

The test procedure with each unit was exactly the same and in general it may be divided into two parts; (1) that in which the conditions were in equilibrium or steady, and (2) that in which the conditions were unsteady due to the fact that the room was cooling or heating. In each case the unit was installed under the window in the exposed wall of the test room as shown in Fig. 2, and the temperatures in the cold room and attic were maintained constant at about minus 1.0 F and 62.0 F respectively. Whenever steam was supplied

[5] The Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperature, by A. C. Willard, A. P. Kratz, and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 303).

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TABLE 1—DIMENSIONS AND TEMPERATURES OF UNITS, AND ENERGY AVAILABLE AND EMITTED DURING EQUILIBRIUM AND CHANGING CONDITIONS IN FIRST COOLING PERIOD

Unit No.	Convector No.	Description of Unit	Over-all Dimensions of Unit, Inches			Over-all Dimensions of Heating Unit, Inches			Weight in Lb			Volume of Steam Space, Cu Ft	Weight of Saturated Steam in Unit in Lb at 216.5 F	Temperature in Deg F under Equilibrium Conditions with Steam Temperature at 216.5 F	
			Length	Height	Width	Length	Height	Width	Heating Unit	Cabinet	Total			Heating Unit	Cabinet
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
1	..	8-Sec., 26 in., 5-Tube Cast-Iron Radiator[a]	20	26	8	20	26	8	151.0	..	151.0	0.3180	0.0129	214.0	..
2	..	18-Sec., 25 in., 3-Tube Cast-Iron Radiator[b]	27	25	3½	27	25	3½	115.0	..	115.0	0.2935	0.0119	214.5	..
3	32	Convector with Cast-Iron Heating Unit	34½	29	5⅞	33⅜	6	5¾	94.0	36.0	130.0	0.1448	0.0059	213.6	130.0
4	1	Convector with Cast-Iron Heating Unit	37⅝	26	5⅝	36	14³⁄₁₆	5⅜	82.0	36.0	118.0	0.1463	0.0059	213.3	124.1
5	38	Convector with Non-Ferrous Heating Unit	36½	29	5⅜	35½	5¼	5	33.0	38.0	71.0	0.0508	0.0021	213.6	120.1
6	6	Convector with Non-Ferrous Heating Unit	30	28	5⅝	29⅝	2	5¼	13.0	30.0	43.0	0.0162	0.0007	212.7	112.3

[a] Sections on 2½-in. centers.
[b] Sections on 1½-in. centers.

TABLE 1 (*Continued*)—DIMENSIONS AND TEMPERATURES OF UNITS, AND ENERGY AVAILABLE AND EMITTED DURING EQUILIBRIUM AND CHANGING CONDITIONS IN FIRST COOLING PERIOD

Unit No.	Convector No.	Description of Unit	Heat Emitted to Room Under Equilibrium Conditions, Btu (Temp. of Steam=216.5 F)				Heat Output of Units Based on Standard Equilibrium Conditions			Approximate Thermal Energy Available in Unit Above 60 Deg F When Steam Supply Was Turned Off, Btu				
			From Heating Unit per Hour	From Eupatheoscope per Hour	Total		Sq Ft of Equivalent Direct Radiation (E D R)	Weight of Metal, Lb per Sq Ft of Equivalent Direct Radiation		In Metal of Heating Unit	In Metal of Cabinet	Total in Metal of Heating Unit and Cabinet	In Steam in Heating Unit	Total in All Metal and Steam
					Per Hour	Per 10 Min		Based on Heating Unit	Based on Heating Unit and Cabinet					
1	2	3	17	18	19	20	21	22	23	24	25	26	27	28
1	..	8-Sec., 26 in., 5-Tube Cast-Iron Radiator[a]	6308	68	6376	1063	28.1	5.37	5.37	2768	...	2768	15	2783
2	..	18-Sec., 25 in., 3-Tube Cast-Iron Radiator[b]	6463	68	6531	1089	28.3	4.06	4.06	2114	...	2114	13	2127
3	32	Convector with Cast-Iron Heating Unit	6192	77	6269	1045	25.1	3.74	5.18	1718	285	2003	7	2010
4	1	Convector with Cast-Iron Heating Unit	5833	78	5911	985	23.8	3.44	4.96	1496	261	1757	7	1764
5	38	Convector with Non-Ferrous Heating Unit	6173	77	6250	1042	25.0	1.32	2.84	537	258	795	2	797
6	6	Convector with Non-Ferrous Heating Unit	5959	88	6047	1008	23.8	0.55	1.81	210	177	387	1	388

[a] Sections on 2½-in. centers.
[b] Sections on 1½-in. centers.

TABLE 1 (*Continued*)—DIMENSIONS AND TEMPERATURES OF UNITS, AND ENERGY AVAILABLE AND EMITTED DURING EQUILIBRIUM AND CHANGING CONDITIONS IN FIRST COOLING PERIOD

UNIT No.	CONVECTOR No.	DESCRIPTION OF UNIT	TEMPERATURES IN DEG F 10 MIN AFTER STEAM SUPPLY WAS TURNED OFF		APPROXIMATE HEAT EMITTED TO ROOM DURING THE 10 MIN PERIOD FOLLOWING THE TURNING OFF OF THE STEAM SUPPLY, BTU						TEMP. DIFF. BETWEEN FLOOR AND CEILING 10 MIN AFTER STEAM SUPPLY WAS TURNED OFF
			Heating Unit	Cabinet	From Metal in Heating Unit	From Metal in Cabinet	Total from Metal in Heating Unit and Cabinet	From Steam in Unit	From Eupatheoscope	Gross Total	
1	2	3	29	30	31	32	33	34	35	36	37
1	..	8-Sec., 26 in., 5-Tube Cast-Iron Radiator[a]	167.7	....	832	....	832	13	13	858	19.5
2	..	18-Sec., 25 in., 3-Tube Cast-Iron Radiator[b]	157.7	....	777	....	777	12	13	802	18.7
3	32	Convector with Cast-Iron Heating Unit	154.6	112.0	660	73	733	6	14	753	19.1
4	1	Convector with Cast-Iron Heating Unit	147.0	108.5	647	63	710	6	15	731	17.4
5	38	Convector with Non-Ferrous Heating Unit	81.7	87.3	461	141	602	2	15	619	13.8
6	6	Convector with Non-Ferrous Heating Unit	66.8	81.3	201	105	306	1	18	325	15.3

[a] Sections on 2½-in. centers.
[b] Sections on 1½-in. centers.

to the unit, the steam temperature was held constant at 216.5 F. The temperature maintained in the basement depended upon the air temperature 3 in. above the floor as established by each unit, and in each case it was regulated so that there was practically no heat loss or gain through the floor. Although no attempt was made to regulate these temperatures, external to the test room, during the portions of the tests over which the test room was heating up or cooling down, it was found that they did not vary an appreciable amount during these comparatively short periods of unsteady operation.

The surface temperature of the eupatheoscope cylinder was maintained at 83 F and the equivalent temperature of the environment in the test room was obtained for all operating conditions.

At the end of from 24 to 48 hours after a unit was installed in the test room, the air and wall temperatures were in complete equilibrium, and under these conditions of steady operation the following temperatures were observed and recorded: air temperatures in the cold room, attic and basement; air temperatures in the center of the test room at distances 3 in., 30 in., and 60 in. above the floor and 3 in. below the ceiling; floor, ceiling and wall surface temperatures; and the temperature of the steam at the entrance to the unit. In the case of the radiators, the approximate temperature of the radiator itself was obtained by means of a thermocouple soldered to the surface of the center section, and in the case of the convectors, the approximate temperatures of the prime heating surface and the cabinet were obtained in the same manner. Simultaneously with these temperature observations, the net amount of steam condensed by the unit was determined at 10-min intervals over a period of one hour.

At the end of the equilibrium portion of the test the steam supply to the unit was turned off at the control valve on the lower floor of the laboratory, and successive sets of the temperatures previously enumerated were observed as rapidly as possible during the entire time that the room was cooling from the temperature existing at equilibrium. These observations were continued until the temperature at the 30-in. level reached 60 F. Care was taken to record the elapsed time from the beginning of the cooling period with each set of temperature observations.

When the temperature at the 30-in. level reached a minimum of 60 F the steam supply was turned on and again successive sets of temperature observations were made until the air temperature at the 30-in. level reached 68 F. The amounts of steam condensed in the unit over small intervals of time were obtained during the entire heating period for the purpose of determining the variation in condensing rate from the time that the steam supply was turned on until equilibrium conditions were practically reached.

A second cooling period, starting at the time that the steam supply was turned off when the temperature at the 30-in. level had reached 68 F, continued until the temperature had again dropped to 60 F. The temperature observations made during this cooling period were the same as those made during the first cooling period. The major difference in the two periods was in the degree of thermal equilibrium existing between the unit, the air, and the building materials in the walls, floor and ceiling of the room at the start of the period. In the first case the room and unit had been operating under steady conditions for a number of hours and the temperature at the 30-in.

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TABLE 2—TEMPERATURE OF AIR AND INSIDE SURFACES OF EXPOSED WALLS UNDER EQUILIBRIUM CONDITIONS

Unit No.	Description of Unit	Temperature of Air and Inside Surfaces of Exposed Walls Under Equilibrium Conditions, F										Difference in Temperature, F, Between Air at Center of Room and Inside Surfaces of Walls at the Same Level			Difference[a] Between Aver. of Temp. of Air at Center of Room and Aver. of Temp. of Inside Surfaces of Exposed Walls, F
		Air in Cold Room	Air at Center of Room					Inside Surface of Walls							
			3-in. Above Floor	30-in. Level	60-in. Level	3-in. Below Ceiling	Difference Between Floor and Ceiling, F	30-in. Level	60-in. Level	3-in. Below Ceiling	Back of Unit	30-in. Level	60-in. Level	3-in. Below Ceiling	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
1	8-Sec., 26 in., 5-Tube Cast-Iron Radiator[b]	−1.0	63.1	71.2	78.5	84.5	21.4	58.7	61.9	69.1	128.5	12.5	16.6	15.4	14.8
2	18-Sec., 25 in., 3-Tube Cast-Iron Radiator[c]	−1.2	62.1	71.1	77.0	83.2	21.1	58.4	61.3	68.6	136.4	12.7	15.7	14.6	14.3
3	Convector with Cast-Iron Heating Unit	−1.1	60.9	70.8	79.1	84.8	23.9	58.1	61.7	68.5	82.6	12.7	17.4	16.3	15.5
4	Convector with Cast-Iron Heating Unit	−1.6	61.1	71.6	77.7	82.1	21.0	57.9	60.4	66.9	72.1	13.7	17.3	15.2	15.4
5	Convector with Non-Ferrous Heating Unit	−1.3	61.0	71.1	79.4	84.3	23.3	58.2	61.7	68.4	76.0	12.9	17.7	15.9	15.5
6	Convector with Non-Ferrous Heating Unit	−0.9	59.9	70.2	78.5	83.4	23.5	57.6	60.9	68.2	74.3	12.7	17.6	15.2	15.2

[a] Based on temperatures at 30-in. level, 60-in. level and 3-in. below ceiling.
[b] Sections on 2½-in. centers.
[c] Sections on 1½-in. centers.

level was approximately 71.0 F, while in the second case the cooling period was preceded by two periods of unsteady operation, one with the room cooling, and one with the room heating, and the temperature at the 30-in. level was about 68 F. That is, the first period was started under conditions of complete thermal equilibrium, while at the start of the second period such thermal equilibrium did not exist.

Following the second cooling period the steam supply to the unit was again turned on and for the remainder of the day and the following night the test plant, including the test room, was operated under steady conditions with the temperatures in the cold room, attic, and basement maintained constant at the values previously stated. The same routine of test, consisting of two cooling-down periods and one heating-up period, was then repeated for at least the second and sometimes the third day.

RESULTS OF TEST

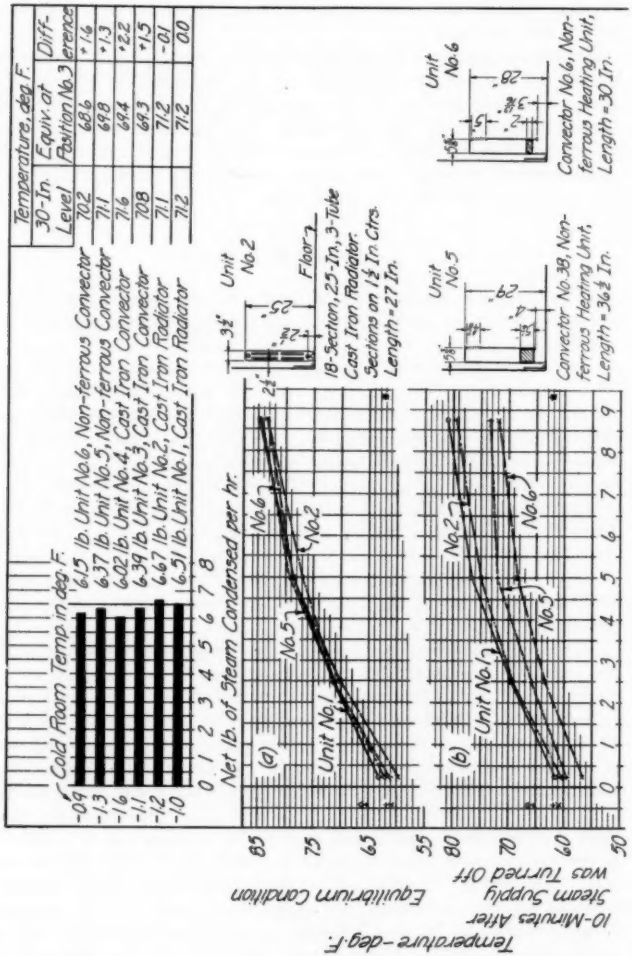
In general the results may be discussed by considering the performance of the units under (1) equilibrium or steady conditions, and (2) under unsteady conditions.

The temperature gradients or the air temperatures in the test room 3 in., 30 in., and 60 in. above the floor and 3 in. below the ceiling are shown in Figs. 4a and 4c for the six different units tested under equilibrium conditions. The numerical values of the difference in temperature between the floor and ceiling obtained with the different units are given in col. 8, Table 2. Although the sizes or capacities of the units were limited to those commercially available, the temperatures at the 30-in. level with five of the units came within the range of from 70.8 to 71.6 F, a variation of less than one degree. The temperature with unit No. 6, which was a convector with a non-ferrous heating unit, was 70.2 F, a variation from the maximum 30-in. level temperature of 1.4 F. Thus, with the exception of the latter unit, the temperatures at the 30-in. level were practically the same.

For the purpose of comparison, the temperature gradients for unit No. 1, the cast-iron radiator with the sections on 2½-in. centers, are given in both Figs. 4a and 4c. The temperature gradient curves for units Nos. 3 and 4, the two convectors with cast-iron heating units, are shown in Fig. 4c. It may be observed that, with the same temperature at the 30-in. level, the temperature at the floor with the two convectors was about 2.0 F lower than it was with the radiator. Unit No. 4, as a result of a lower temperature at the ceiling, gave a slightly less over-all difference in temperature between the floor and ceiling than that obtained with either unit No. 1 or unit No. 3. Fig. 4a, in addition to the temperature gradient curve for unit No. 1, includes the curves for units Nos. 2, 5, and 6. Unit No. 2 was a comparatively light weight radiator with sections on 1½-in. centers, and units Nos. 5 and 6 were convectors with non-ferrous heating units. As indicated by these curves and the data in col. 8, Table 2, the floor to ceiling temperature differences for the two radiators were approximately the same, but were slightly less than those for the two non-ferrous convectors, which were also practically alike.

Following the completion of each equilibrium test, the steam supply to the unit was turned off and the test room was allowed to cool until the temperature at the 30-in. level reached 60 F. Figs. 4b and 4d show the temperature

gradients in the test room for each unit 10 min after the steam supply was turned off. These curves, obtained while the room was cooling, compared with the curves in Figs. 4a and 4c, obtained under equilibrium conditions, show


Temperature, deg. F.
30-In. Level
Equiv. at Position No. 3
Difference
702 68.6 +1.6
711 69.8 +1.3
716 69.4 +2.2
708 69.3 +1.5
711 71.2 -0.1
712 71.2 0.0
6.15 lb. Unit No. 6, Non-ferrous Convector
6.37 lb. Unit No. 5, Non-ferrous Convector
6.02 lb. Unit No. 4, Cast Iron Convector
6.39 lb. Unit No. 3, Cast Iron Convector
6.67 lb. Unit No. 2, Cast Iron Radiator
6.51 lb. Unit No. 1, Cast Iron Radiator
Cold Room Temp. in deg. F.
-0.9
-1.3
-1.6
-1.1
-1.2
-1.0
0 1 2 3 4 5 6 7 8
Net lb. of Steam Condensed per hr.
Unit No. 2
Floor
18-Section, 25-In, 3-Tube Cast Iron Radiator. Sections on 1¼ In. Ctrs. Length = 27 In.
Unit No. 5
Convector No. 38, Non-ferrous Heating Unit. Length = 36½ In.
Unit No. 6
Convector No. 6, Non-ferrous Heating Unit. Length = 30 In.
(a)
(b)
Unit No. 1
No. 2
No. 3
No. 5
No. 6
85 75 65 55
80 70 60 50
Temperature - deg. F.
10-Minutes After Steam Supply was Turned Off
Equilibrium Condition

that the temperature difference between the floor and the ceiling decreased when the supply of steam to the heating unit was turned off. A comparison of the numerical values of the temperature differences given in col. 8, Table 2, and col. 37, Table 1, shows that the average decrease in the temperature dif-

ference at the end of 10 min after the steam supply was turned off was 2.2 F for the radiators, 4.2 F for the convectors with the cast-iron heating units, and 8.9 F for the convectors with the non-ferrous heating units.

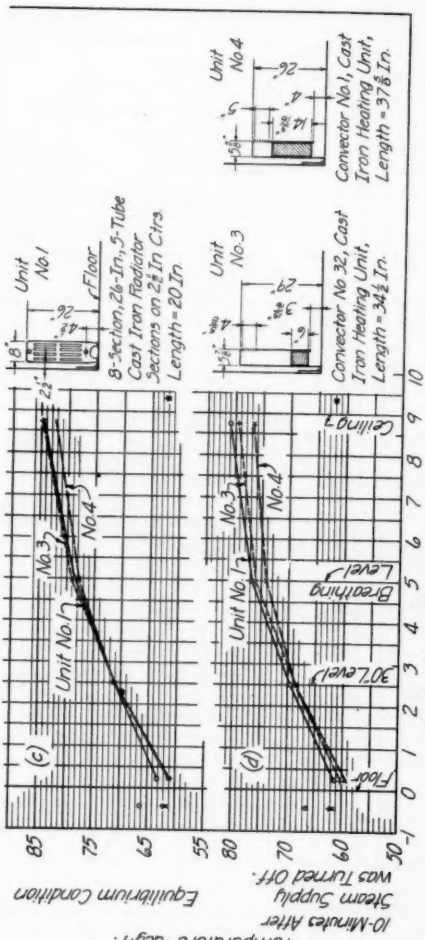
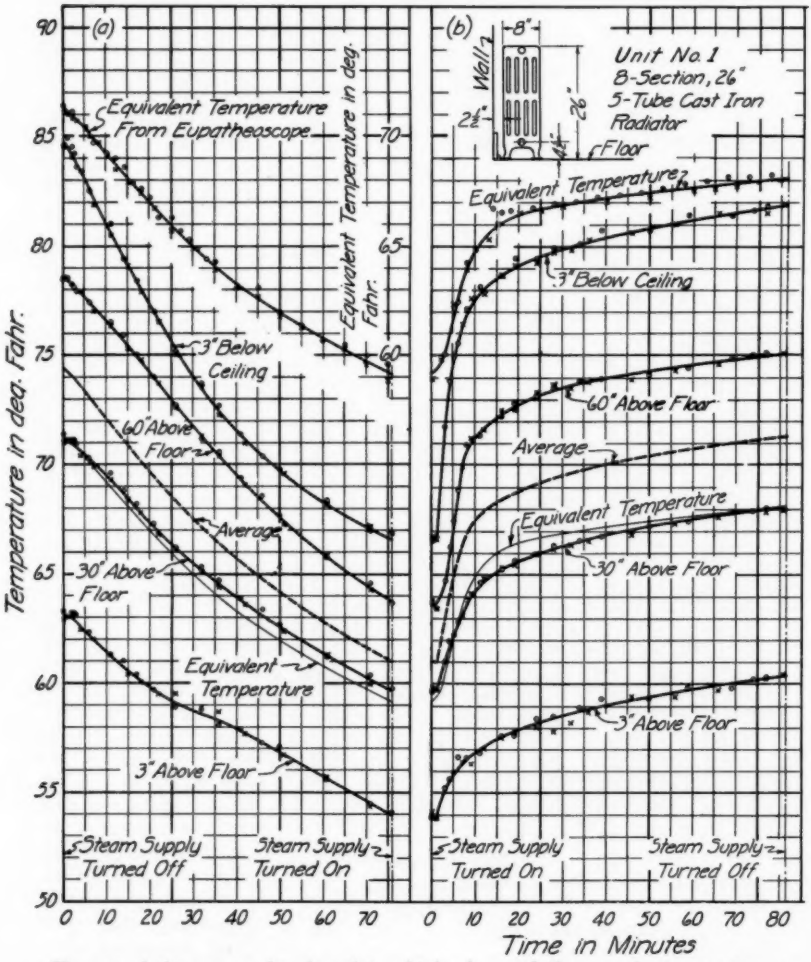


FIG. 4. ROOM TEMPERATURE GRADIENTS FOR EQUILIBRIUM CONDITIONS AND FOR CHANGING CONDITIONS 10 MIN AFTER STEAM SUPPLY WAS TURNED OFF

The chart and the accompanying data at the top of Fig. 4 show the net amount of steam condensed by each unit in pounds per hour under equilibrium conditions and the temperatures which existed during the tests. The small table gives the dry-bulb temperature at the 30-in. level, the equivalent temperature as determined by the eupatheoscope, and the difference between the two temperatures. The capacities of the units in square feet of equivalent direct radiation (EDR) based on the net amount of steam condensed corrected to standard conditions with respect to steam and air temperatures and on a heat output of 240 Btu per square foot per hour are given in col. 21, Table 1. A comparison of the capacities of units Nos. 1 to 5 inclusive shows that when direct radiators were used the capacities required to heat the room to 71 F at the 30-in. level were slightly larger than they were when convectors were used. This apparent advantage of the convectors is offset by the fact that, although the dry-bulb temperatures produced by the units were the same, the equivalent temperatures obtained with the convectors were lower than those obtained with the direct radiators. The correlation between human comfort, dry-bulb temperature, and

equivalent temperature has been discussed in a previous paper [6] in the 1933 TRANSACTIONS, in which it was shown that, if both radiators and convectors were of the proper size to maintain the same equivalent temperatures, the differences in the capacities based on the actual amounts of steam condensed were negligible.



Characteristic curves showing data obtained as relating to air temperatures and equivalent temperatures in the test room during the tests conducted with units Nos. 1, 4, and 5 are given in Figs. 5, 6, and 7. As these curves are representative of the performance of the three types of units tested, similar data

[6] Loc. Cit. See Note 5.

obtained with the remaining radiators and convectors are not shown, except in Figs. 8 and 9 in which all performance curves are assembled in order to facilitate comparison. Referring to Figs. 5, 6, and 7, it may be noted that each figure has been divided into three parts, *a*, *b*, and *c*. In each case, *a* pertains to the first cooling period when steam supply to the unit was turned off at completion of the equilibrium portion of the test and the room was allowed to cool until the temperature at the 30-in. level reached 60 F; part *b* pertains to the heating period immediately following the first cooling period; and part *c* pertains to the second cooling period, which started at the end of the heating period at the time that the steam supply was turned off when the temperature at the 30-in. level reached 68 F. Curves showing the equivalent temperature in the test room are also included for all periods of the tests.

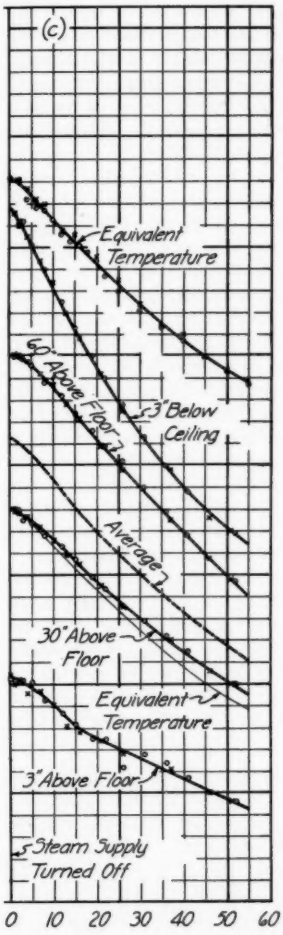


FIG. 5. COOLING AND HEATING CURVES FOR ROOM WITH UNIT NO 1; 8-SECTION, 26-IN., 5-TUBE CAST-IRON RADIATOR

Referring to Figs. 5a, 6a, and 7a it may be observed that during the first portion of the cooling periods the temperatures of the air 3 in. below the ceiling decreased more rapidly than those at any of the other four levels in

the room. In the case of the cast-iron radiator, unit No. 1, shown in Fig. 5a, the cooling near the ceiling continued at a rate greater than that at the other levels until the temperature at the 30-in. level reached approximately 63 F. This occurred about 45 min after the steam supply was turned off. From that time on until the end of the cooling period, the temperature of the air near the ceiling decreased more slowly than it did at the other levels in the room. Likewise, with units Nos. 4 and 5, Figs. 6a and 7a, the temperature of the air near the ceiling decreased more rapidly than it did at the other levels immediately following the start of the cooling period. As the cooling progressed, the rate at which the upper portion of the room cooled changed so materially that in both cases, during the latter part of the cooling period, the relationship between the rates of cooling in the upper and lower portions of the room was the reverse of what it was during the first part of the period. This change in the relative cooling rates of the two parts of the room, as shown by the temperature of the air 3 in. below the ceiling in Figs. 6a and 7a, occurred about 27 min after the steam supply was turned off from unit No. 4 and only 8 min after it was turned off from unit No. 5. An examination of the temperatures of the heating units at the time that the rate of cooling near the ceiling ceased to be greater than the rates at the other levels in the room, showed that in each case those temperatures were less than 100 F. This would seem to indicate that at this time the major portion of the thermal energy which was stored in the units when the steam supply was turned off had been dissipated into the room. No doubt the relatively high air temperatures occurring near the ceiling at the beginning of the cooling periods were brought about by heated air emitted directly from the units, and it is reasonable to assume that these temperatures would be the first to be affected when the steam supply was turned off. It is also reasonable to assume that the thermal energy stored in the units would have an appreciable effect on the room temperature gradients during the time that the stored energy was being transmitted to the room in quantities comparable to the heat loss from the room. As this supply of energy decreased, a point would be reached at which it no longer had an appreciable influence on the temperature gradient in the test room, and a readjustment of the relative rates of cooling in the room would probably occur. From then on the room would probably cool at a more uniform rate at all levels. This may serve as an explanation for the uniformity in the cooling rates that existed during the latter part of the cooling periods for the radiator and the two convectors, as shown in Figs. 5, 6, and 7.

The temperatures of the air at the four levels in the test room during the heating periods are given by the curves in Figs. 5b, 6b, and 7b. In each case, during the first part of the heating period, the temperatures near the ceiling increased more rapidly than those at the lower levels. Also, in every case, the rate of increase in the temperatures at all levels during the first part of the heating period was equal to or greater than the rates of cooling at the corresponding level during the first part of the cooling period. These comparatively high heating rates continued for about 10 min after the steam supply was turned on, and at the end of that time there was a tendency for the temperature rise to be more gradual and uniform at all levels in the room.

Thus, during the first part of both the cooling and heating periods, with both the radiator and the convectors, the rate at which the temperature of the air

decreased or increased, varied at different levels in the room. The maximum rate of change occurred near the ceiling and this rate decreased progressively with the level in the room to a minimum occurring in the level near the floor. Also, as the cooling or heating progressed, the rates at which the temperature of the air was changing at different levels in the room became more nearly the same, so that during the latter part of each cooling or heating period, with both the radiator and the convectors, the air was changing in temperature at approximately the same rate at all levels in the room.

Curves giving the temperature of the air at the different levels in the room during the second cooling period are shown in Figs. 5c, 6c, and 7c. This cooling period differed from the first in that it was preceded by a period in which the conditions were changing. These tests were included because the changing conditions would more nearly stimulate practical conditions present under intermittent operation with thermostatic control. In general, the result curves fall in the same order and appear to have the same relationship to each other as they did during the first cooling period, and apparently the difference in the room and air conditions at the start of the two cooling periods did not materially affect the manner in which the room cooled. It is, therefore, assumed that the discussion devoted to the results presented for the first cooling period is also applicable to the results for the second cooling period.

The curves shown as broken lines in Figs. 5, 6, and 7 give the average air temperatures at the center of the test room. In each case they represent an average of the four full line curves which give the air temperatures 3 in., 30 in., and 60 in. above the floor and 3 in. below the ceiling. With all three units the average temperature at the start of the first cooling period fell between the temperatures at the 30-in. and the 60-in. levels. In the case of the radiator, Fig. 5a, it was more nearly midway between these two temperatures than it was in the case of the two convectors, as shown in Figs. 6a and 7a. Referring to Figs. 4a and 4c, the relatively low floor and ceiling temperatures maintained with unit No. 4, and the relatively low floor temperature maintained with unit No. 5 serve to explain why the average temperature for these two convectors was relatively lower than it was for the radiator. Since the average temperature in the room with both the radiator and the convectors was always lower than that at the 60-in. level, it is evident that calculated heat losses from the room based on the temperature at the 60-in. level will tend to be too high, and the amount of the error will tend to be larger with the convectors than with the radiator.

In order to facilitate comparison between the equivalent temperature and the dry-bulb temperatures at the center of the room, the original equivalent temperature curves have been superimposed on the dry-bulb temperature scales and are shown as light-weight full lines in Figs. 5, 6, and 7. It may be observed that in all cases these equivalent temperature curves lie near the curves giving the temperature at the 30-in. level. At the start of the first cooling period, with the radiator, the equivalent and dry-bulb air temperatures at the 30-in. level were the same, while with both convectors the equivalent temperatures were about 1.5 F lower than the dry-bulb temperatures. This difference was previously mentioned under the discussion of the temperature gradient curves shown in Figs. 4a and 4c. As the cooling progressed in the case of the raditor, the equivalent temperature decreased at a relatively greater rate than that for the 30-in. level temperature so that during the latter part

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of the cooling period it was slightly lower than the latter temperature. In the case of the convectors, during the cooling period the difference between the equivalent and the 30-in. level temperatures decreased a small amount over that which existed at the beginning of the period. During the first part of the heating period, with both the radiator and the convectors, the equivalent temperature rose rapidly to a value above or at least equal to the 30-in. level temperature. However, as the heating period continued, the equivalent temperatures with the convectors became less than the 30-in. level temperatures and, with the radiator, the two temperatures became the same.

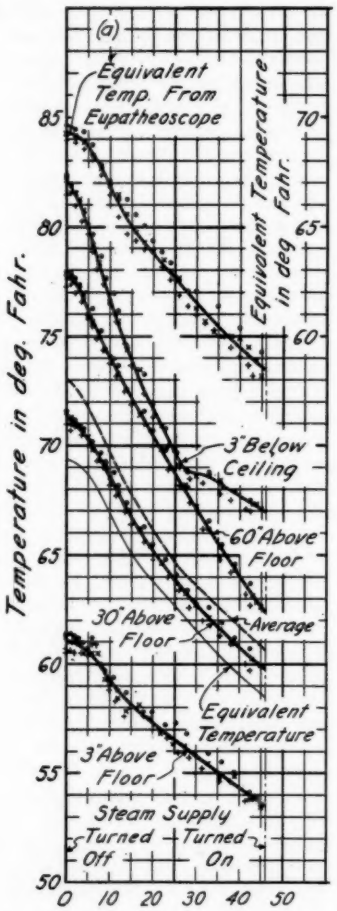


FIG. 6. COOLING AND HEATING CURVES FOR ROOM WITH UNIT NO. 4; CONVECTOR WITH CAST-IRON HEATING UNIT

In Figs. 8 and 9 curves showing the temperatures obtained with all six of the units tested have been assembled. The temperatures 3 in. and 30 in. above

the floor and 3 in. below the ceiling for the two cooling periods and the one heating period are shown in Fig. 8, while the equivalent temperatures and the average of the temperatures at the four levels in the room are included in

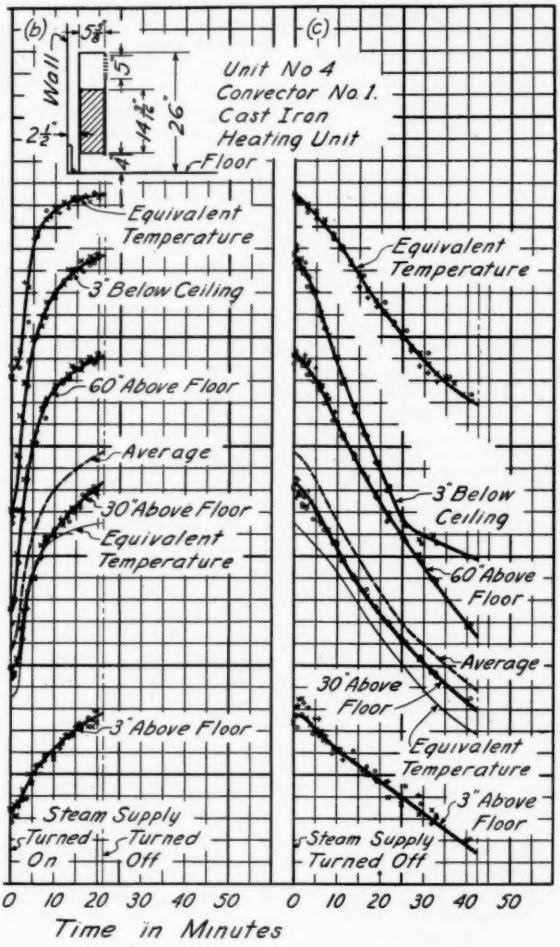


Fig. 9. In every case the results obtained with the direct cast-iron radiators, units Nos. 1 and 2, are represented by full lines, those obtained with the convectors having cast-iron heating units, units Nos. 3 and 4, are represented by dashed lines, and those obtained with convectors having non-ferrous heating units, units Nos. 5 and 6, are represented by dotted lines.

Referring to Fig 8a, which gives the results obtained during the first cooling period, it may be observed that with the exception of unit No. 6, the temperatures at the 30-in. level at the beginning of the period were practically the same with all of the units. Also, from this group of curves it may be noted that when

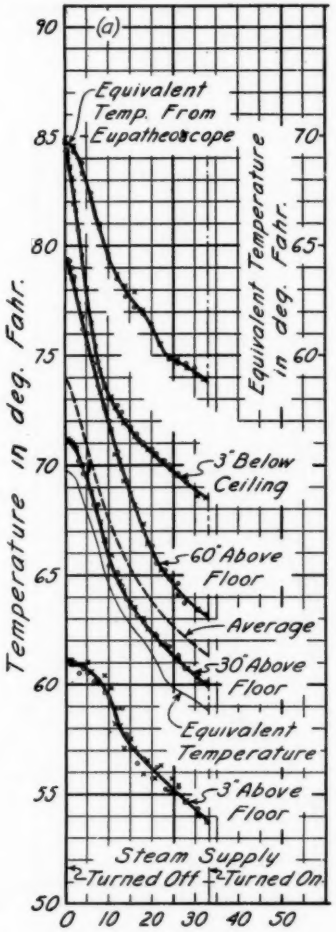
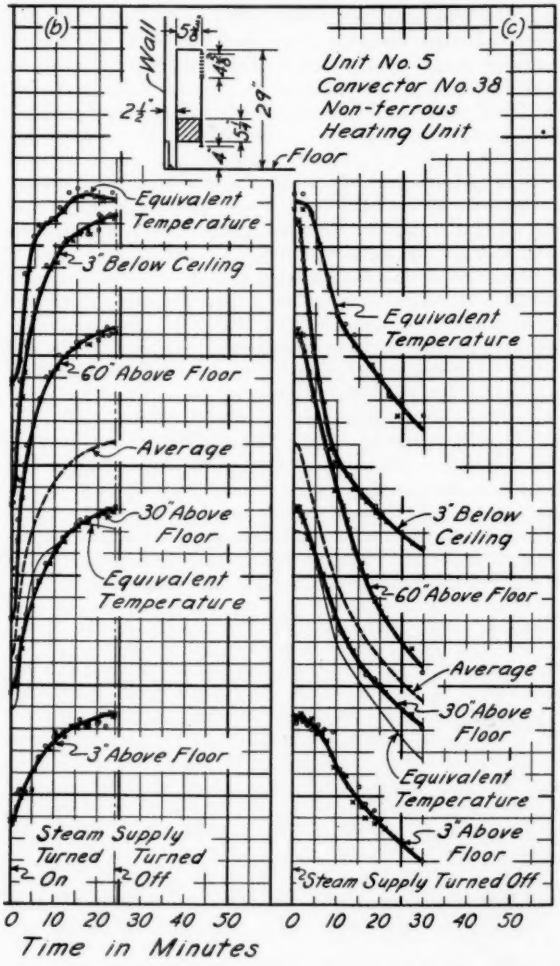


FIG. 7. COOLING AND HEATING CURVES FOR ROOM WITH UNIT NO. 5; CONVECTOR WITH NON-FERROUS HEATING UNIT

the steam supply was turned off, the rate at which the room cooled at the 30-in. level was slightly different with the two convectors, units Nos. 5 and 6, which had non-ferrous heating units, and was practically the same with the two convectors, units Nos. 3 and 4, which had cast-iron heating units. The cooling rates with the two direct radiators were not the same, but they were both less than the rates obtained with any of the convectors. The upper and lower groups of curves in Fig. 8a give the temperatures

near the ceiling and near the floor for all of the units tested, and it may be observed that in both of these groups of curves the relative position of each curve is the same as it is in the center group, which gives the temperature at the 30-in.



level. Likewise, in Fig. 9a, which gives curves showing the equivalent temperature and the average of the temperatures of the air at four levels in the center of the room for each unit during the first cooling period, the magnitudes of the slopes of the curves for the different units are related to one another in the same way as those shown in Fig. 8a.

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While all of the groups of curves in Figs. 8a and 9a show practically the same relationship between the rates of cooling with the different units, the group of curves showing the average of the room temperatures at four levels has been selected for the purpose of correlating the cooling rates with the physical and thermal factors governing the performance of the units and the test room at the beginning of and during the first cooling period. The major factors affecting the rate at which a room cools when the steam supply to the heating unit is turned off may be divided into two general divisions, one pertaining to the room itself, and the other pertaining to the heating units. The first division would include such factors as the available thermal energy stored in the materials in the walls, floor, ceiling, furnishings, and in the air, and the rate at which heat is being lost from the air to the ambient spaces. Indices relating to the magnitude of these factors are the temperatures and masses of the different materials and the differences in temperature between the air in the room and that in the ambient spaces. In Fig. 9a it may be observed that with all of the units, the deviation in the averages of the temperature of the air in the test room at the start of the cooling period was less than 1.5 F. Referring to Fig. 4, under equilibrium conditions, it may be noted that the temperatures in the ambient spaces were approximately the same for all of the tests. Therefore, insofar as it was possible to obtain them, most of the influencing factors pertaining to the room itself were the same for each unit at the start of the first cooling period. Two appreciable exceptions to this similarity in conditions are indicated by the data in cols. 12 and 16, Table 2. Col. 12 gives the temperature of the inside surface of the exposed wall back of each unit under equilibrium conditions and just preceding the first cooling period. A comparison of these temperatures shows that the wall at this particular location was much warmer, and therefore, possessed more thermal energy at the start of the cooling periods with the radiators than it did at the start of the cooling periods with the convectors.

The difference between the average of the temperature of the air in the center of the room at the 30-in. and 60-in. levels, and 3 in. below the ceiling, and the average of the temperatures of the inside surfaces of the two exposed walls taken at the same levels as the air temperatures, are given in col. 16, Table 2. It may be noted from these data that the difference between the temperature of the air in the room and the temperature of the inside surfaces of the wall was always slightly less with the radiators than it was with the convectors. This indicated that for a given air temperature in the room, the temperature of the exposed walls and, therefore, the stored thermal energy, was slightly higher at the beginning of the cooling periods with the radiators than it was at the beginning of the cooling periods with the convectors. No doubt both of the conditions discussed result from the difference in the manner in which heat is emitted from radiators and convectors, and both of them would tend to cause the room to cool more slowly with radiators than with convectors.

The factors pertaining to the units themselves which might influence the rate at which the room cooled consist of the available thermal energies in the masses of metal constituting the units and in the steam in the units at the time the steam supply was turned off. Another contributing factor might consist of the effectiveness of the unit in transmitting the available energy to

the air in the room. A study of the first two factors involved the weight of each unit and the volume of the steam space, which are given in cols. 12 and 13 in Table 1. Unit No. 1, the radiator with the sections on 2½-in. centers, was the heaviest unit tested, and it also contained the largest volume of steam space. The weight of this unit was 151.0 lb and the volume of its steam space was 0.3180 cu ft. Unit No. 6, a convector having a non-ferrous heating unit, was the lightest unit tested, and it contained the smallest volume of steam space. The total weight of this convector, including the heating unit and the cabinet, was only 43.0 lb and the volume of its steam space was only 0.0162 cu ft. Thus the weight of the heaviest unit was about 3.5 times the weight of the lightest, while the ratio of the maximum steam space to the minimum steam space was 19.6. It is interesting to note the wide variation in the weight of the two convectors having non-ferrous heating units, and also to note that, with the cabinets included, units Nos. 3 and 4, the convectors with cast-iron heating units, both weighed slightly more than the direct radiator having the sections on 1½-in. centers. It is possible that the values in col. 23, Table 1, which give the total weight of metal in pounds per square foot of equivalent direct radiation (EDR) have greater significance than the total weight of the units. These values range from 1.81 to 5.37 lb per square foot of equivalent direct radiation (EDR), the minimum and maximum values being those for the lightest and heaviest units respectively.

As previously stated, the temperatures of the heating surfaces were observed by means of a thermocouple attached to a section of prime heating surface. In the case of the radiators in which the entire heating surface was prime surface, this temperature was for all practical purposes representative of the temperature of the entire mass of metal, and there was reasonable justification for its use in calculating the available energy stored in the unit. However, with the complex combinations of prime and secondary heating surfaces composing the heating units of the convectors, it was impossible to arrive at a temperature that was accurately representative of the actual mean temperature of the mass of material in the unit. Doubtless the temperature thus obtained on a portion of the prime heating surface was higher than the mean temperature of the material in the unit and its use in calculating the available energy resulted in values that were somewhat too large. Similarly, the determination of an accurate mean temperature for the metal in the cabinets of the convectors involved refinements that were regarded as unwarrantd for the purposes of these tests. Therefore, the temperatures given in cols. 15 and 16, Table 1, while correct for the portion of the units where the thermocouples were attached, are, particularly in the case of the convectors, only approximations of the mean temperatures of the masses of metal in the entire units.

The approximate thermal energies available above 60 F in the metal in each unit and cabinet at the beginning of the first cooling period are given in cols. 24, 25, and 26, Table 1. These values were calculated by using the masses of material in the respective units, the specific heats of the metals, and the approximate temperatures given in cols. 15 and 16. They are, therefore, subject to the same limitations as the temperatures involved. The energy in the steam contained in each unit at the time the supply was turned off is given in col. 27, and the approximate total energy is shown in col. 28. These data show that while the quantities of energy contained in the steam in the different units varied over a range of from 15 to 1, the actual magnitudes of the quantities

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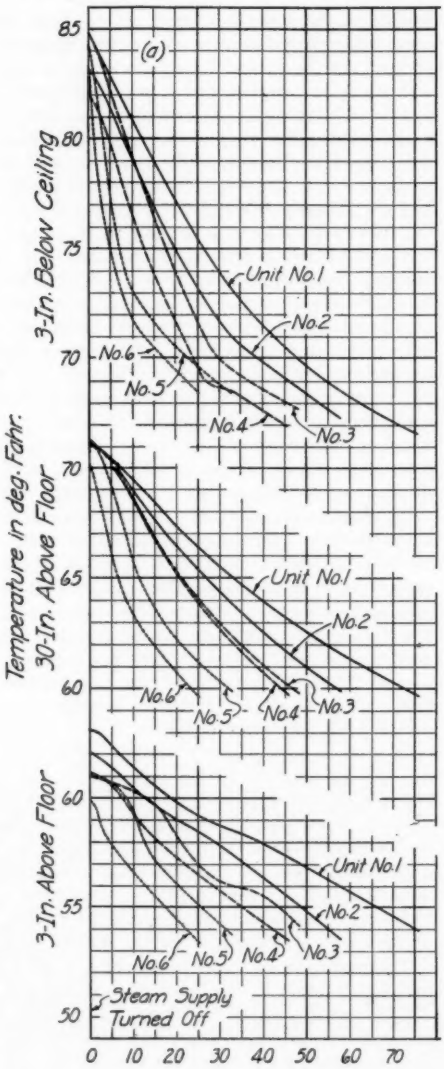
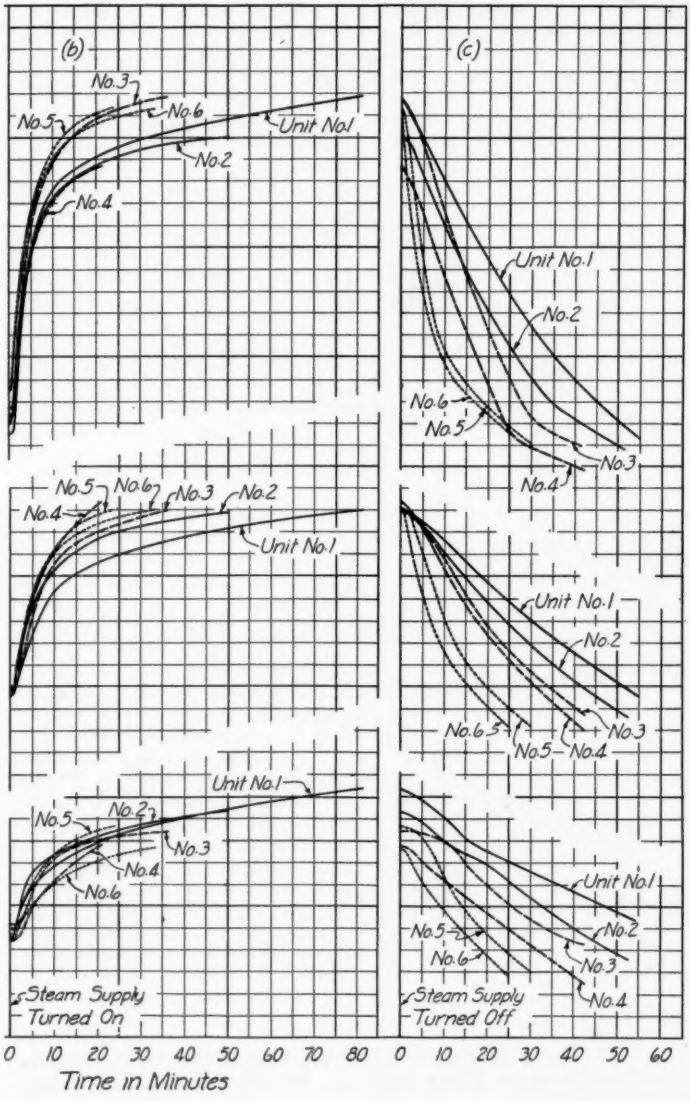


FIG. 8. COOLING AND HEATING CURVES FOR ROOM AT 3-IN. ABOVE FLOOR, 30-IN. ABOVE FLOOR AND 3-IN. BELOW CEILING

involved were so small in comparison with those contained in the metal of the units that the effect on the cooling rates of the room was probably negligible. Therefore, it appears that of the factors pertaining to the units themselves, the available energy stored in the metal of the heating units at the time the steam supply was turned off was the one which had the greatest effect on the rate at which the room cooled.

In correlating the different rates at which the room cooled with the different quantities of available energy in the various units at the time that the steam supply was turned off, the values given in col. 24, Table 1, are probably the most



logical ones to use. These values include only the energy stored in the heating units and in using them the thermal energy stored in the cabinets of the convectors is neglected. This partly compensates for the fact that in the case

of the convectors, the calculated available thermal energies stored in the heating units were somewhat too large, as previously explained. Referring to cols. 10 and 24, it may be noted that in each instance the magnitudes of the calculated available energies fall in the same relative order as the weights

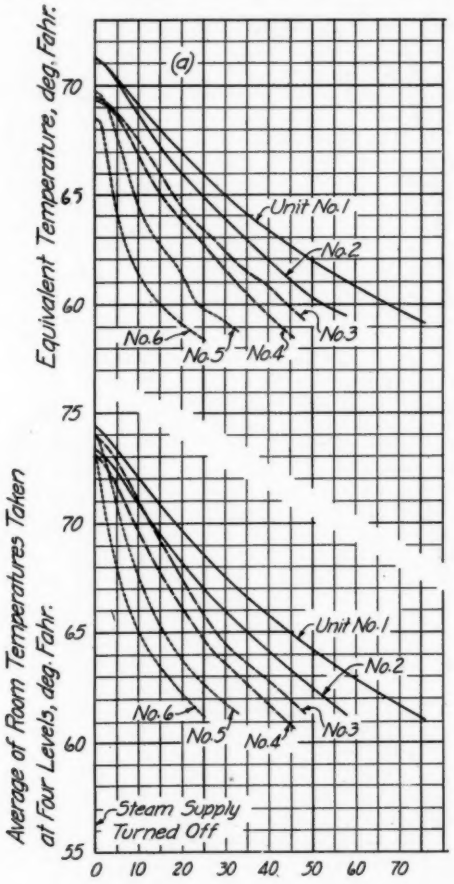
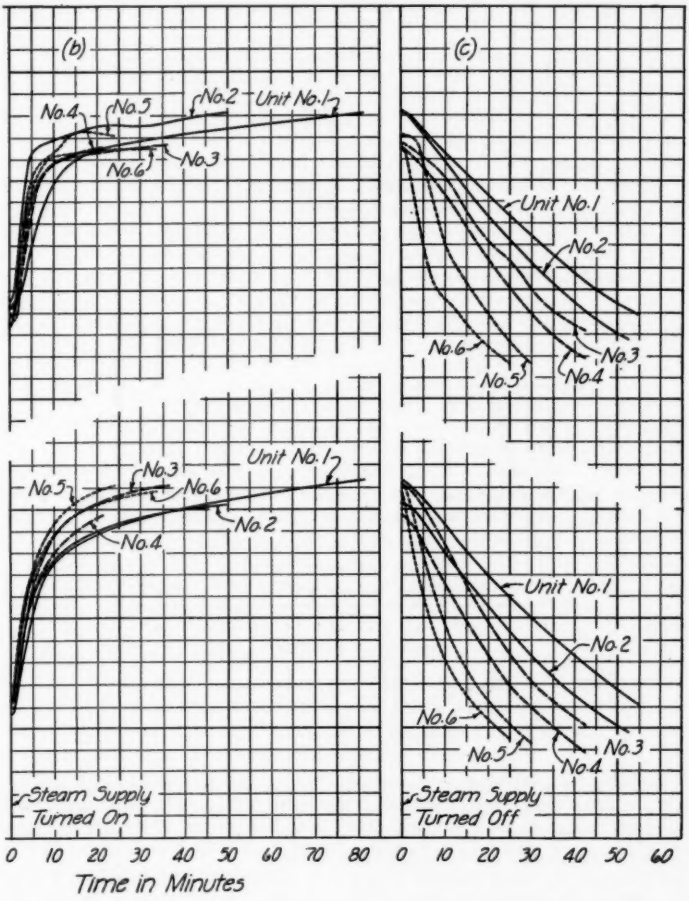


FIG. 9. EQUIVALENT TEMPERATURE AND AVERAGE OF ROOM TEMPERATURES TAKEN AT FOUR LEVELS

of the units. The curves in Fig. 9a which give the average of the temperatures of the air in the room at four levels, are probably the most representative of the conditions in the room at the beginning of and during the first cooling period. These curves exhibit the same characteristics as those for the 30-in. level in Fig. 8a. The latter, however, since they start at the same temperature, afford a more direct means for comparing the slopes of the individual curves.

Differences in cooling rates may be observed by comparing these slopes. The curves for the two radiators show that the rate of cooling with unit No. 1, the heavier of the two radiators, was slightly less than it was for unit No. 2. The curves for the two convectors having cast-iron heating units indicate that



there was practically no difference in the cooling rates with the two units, although the heating unit for No. 4 was slightly heavier than the one for unit No. 3. The curves for the two convectors having non-ferrous heating units show that the cooling rate with unit No. 6 was slightly greater than it was with unit No. 5, especially during the first portion of the cooling period. As the heating unit of unit No. 6 was considerably lighter than the heating unit

of No. 5, the larger cooling rate was obtained with the lightest unit. On comparing the curves for units Nos. 3 and 4, with those for units Nos. 5 and 6, it may be seen that the cooling rates with the lighter types of convectors were greater than they were with the heavier types. Likewise, the cooling rates with the radiators, which were heavier than any of the convectors, were less than the cooling rates with the convectors. Thus, it is apparent that the mass

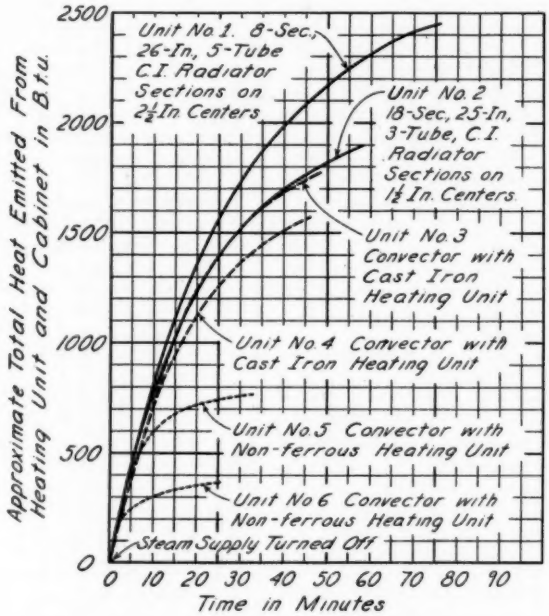

Approximate Total Heat Emitted From Heating Unit and Cabinet in B.t.u.
2500
2000
1500
1000
500
0
Unit No. 1. 8-Sec., 26-In., 5-Tube C.I. Radiator Sections on 2½ In. Centers
Unit No. 2 18-Sec., 25-In., 3-Tube, C.I. Radiator Sections on 1½ In. Centers
Unit No. 3 Convector with Cast Iron Heating Unit
Unit No. 4 Convector with Cast Iron Heating Unit
Unit No. 5 Convector with Non-ferrous Heating Unit
Unit No. 6 Convector with Non-ferrous Heating Unit
Steam Supply Turned Off
0 10 20 30 40 50 60 70 80 90
Time in Minutes

FIG. 10. APPROXIMATE HEAT EMITTED FROM HEATING UNITS AND CABINETS AFTER STEAM SUPPLY WAS TURNED OFF

of metal in a heating unit, which is available for the storage of thermal energy, may become a material factor in affecting the cooling rate of a room when the steam supply is turned off. A comparison of the data given in cols. 20 and 33 of Table 1 still further confirms this conclusion.

Column 20 gives the heat emitted to the test room over a period of 10 min. under equilibrium conditions, and since equilibrium exists during this time, these values also represent the heat loss from the room. Column 33 gives the approximate heat emitted to the room from the various units, during the 10-min period following the turning off of the steam supply, and a comparison of these quantities with those in col. 20 shows that, particularly in the case of the heavier units, the quantities of heat emitted from the metal in the units during the first part of the cooling period were of sufficient magnitude to tend

to maintain the air temperature in the room. Also, the differences in the quantities of heat emitted from the light and heavy units were large enough to make it reasonable to expect a variation in the cooling rates. Fig. 10 shows the approximate heat emitted from the units at various time intervals during the entire cooling period. In the case of the convectors the approximate heat emitted from the cabinet is included with that emitted from the heating units. The curves were obtained from calculations based on the weights of the heating units and cabinets, the specific heats of the metals, and the temperatures previously discussed. They are, therefore, subject to the limitations inherent in the temperature observations, which, as previously explained, in the case of convectors, were somewhat higher than the actual mean temperature of all the metal in the entire heating units and cabinets. A comparison of the curves for the different units, subsequent to the elapse of 15 min of the cooling period, shows that with one exception there was an appreciable difference in the approximate quantity of thermal energy that had been emitted from the units to the air and surroundings. This one exception was with units Nos. 2 and 3, and it is probable that, since the method of calculation gave results somewhat too great for all of the convectors, the curve for the heaviest convector would overlap that for the lightest radiator which consisted entirely of prime surface and had no cabinet. Furthermore, the difference in weight between units Nos. 2 and 3 was comparatively small.

In Fig. 9c the curves show the equivalent temperature and the average of the temperature of the air in the room at four levels for all of the units during the second cooling period. This cooling period started at the end of the warming period shown in Fig. 9b. These curves exhibited the same characteristics as those shown in Fig. 9a and hence lead to the same conclusions. In general, the results obtained during the two cooling periods indicated that the mass of material and the thermal energy stored in the units at the beginning of the periods had an appreciable effect on the cooling rates. However, any attempt to calculate the temperature of the air in the room or the temperature of the heating unit by the use of equations based on the assumption of equilibrium between the heat emitted from the unit and the instantaneous heat loss from the room indicated that the influence of the heat capacity of the walls, floor, and ceiling of the room was sufficiently great to invalidate the results obtained by such calculations. The calculated temperatures of both the air in the room and of the metal in the heating units gave cooling rates that were much greater than those actually obtained by test.

The results obtained during the heating period are shown in Figs. 8b and 9b. The center group of curves in Fig. 8b shows the rates at which the temperature of the air at the 30-in. level increased when the steam supply was turned on, and the lower and upper groups of curves give the same information with respect to the temperature of the air 3 in. above the floor and 3 in. below the ceiling. From the data in Fig. 8b it may be observed that with each unit the temperature of the air near the ceiling increased at a greater rate than that for the temperature of the air at the 30-in. level. Also, in every case the rate of heating of the air at the 30-in. level was greater than the rate of heating of the air 3 in. above the floor. Referring to the center group of curves in Fig. 8a, it may be noted that under equilibrium conditions, or at the beginning of the cooling period, all of the units, except unit No. 6, were of sufficient size to maintain practically the same temperature at the 30-in. level.

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From the center group of curves in Fig. 8b it may be observed that during the first 8 min of the heating period the heating rates with all of the units, with the exception of unit No. 1, were practically the same. From that time on, the rates with units Nos. 2 and 3 decreased, and during the latter part of the period they became considerably less than the rates with units Nos. 4 and 5. These latter two units, one a convector with a cast-iron heating unit, and one a convector with a non-ferrous heating unit, maintained approximately the same heating rates during the greater portion of the heating period. The heating rate with the heaviest radiator, unit No. 1, was less than the heating rate with all of the other units during the entire period, and that for unit No. 2, the light weight radiator, was less than those with the convectors, after about 8 min of the period had elapsed. While the results based on the rate of heating of the air at the 30-in. level indicated that the heating rates with the radiators were less than those with the convectors, they did not show any consistent difference between the heating rates with the convectors having cast-iron heating units, and those having non-ferrous heating units.

The performances of the units during the heating period may also be compared on the basis of the average of the temperatures of the air at four levels in the room. These data are given in Fig. 9b, and from them it may be observed that on this basis the heating rates with the two radiators were approximately the same. The heating rates with all of the convectors were greater than those with the radiators, and although the difference between the rates with the various convectors was not great, the results indicated that the rates with the convectors having non-ferrous heating units were somewhat greater than those with the convectors having cast-iron heating units.

In general, the differences between the performances of the various units during the heating period were not as great nor as clearly marked as they were during the cooling periods. This, no doubt, was due to the fact that, within 5 min after the steam was turned on at the start of the heating period, the air in the steam space was all expelled and the heating surfaces attained full operating temperatures. From that time on the heating of the room was independent of the masses of the units, and was dependent upon the capacities of the units and the effectiveness with which they emitted heat to the air and the surroundings. Inasmuch as the capacities of the convectors were practically the same, any small differences in the rates at which the room heated would be explained by slight differences in the heating effect of the individual convectors. The heating effect of the radiators differed from that of the convectors in that a much larger portion of the heat was given off by direct radiation. This direct radiation heated the walls and surroundings without having an immediate effect on the temperature of the air in the room. Hence, the rate of heating the room, as measured by the rate of temperature rise for the air, was somewhat less for the radiators than it was for the convectors. That considerable effective radiation was present in the room, particularly in the case with the radiator designated as unit No. 2, is shown by the upper group of curves in Fig. 9b. These curves give the equivalent temperatures in the room, as observed with the eupatheoscope, during the heating periods with all of the units. As previously explained, the eupatheoscope evaluates the conditions in a non-uniform environment in terms of the temperature in a uniform environment, taking into account the presence of radiation from hot and cold surfaces. The effectiveness of radiation from the hot surfaces of unit No. 2

on the equivalent temperature, may be observed by comparing the relative position of the curves for unit No. 2 in the two groups of curves in Fig. 9b. From these curves it is evident that while the average of the temperature of the air at four levels in the room with unit No. 2 was lower than it was with the convectors, the equivalent temperature with the radiator was higher than it was for any of the convectors.

SUMMARY AND CONCLUSIONS

The results of this investigation indicate that considerable differences may exist between the cooling or heating rates of a room as influenced by different types of heating units. Such results may be useful in the application to problems of temperature control, and particularly to problems arising from the use of different types of heating units in the same heating system.

The following conclusions may be drawn as applying to the conditions under which the tests were conducted:

1. Under equilibrium conditions corresponding with those in winter service, a room can be heated to the same temperature at the 30-in. level with less steam when the latter is used in convectors than when it is used in direct radiators. However, the equivalent temperature produced by convectors is from 1.3 to 2.2 F lower than that produced by direct radiators and any saving in steam consumption is in some measure offset by less comfortable conditions in the room.

2. The rate of cooling of a room is materially influenced by the mass of the heating unit and hence by the thermal energy stored in the unit at the time that the steam is turned off. The lower rates of cooling correspond with the greater masses of the heating units. For the units used in these tests the rate of cooling of the room was greatest with the convectors having non-ferrous heating units, was somewhat less with convectors having cast-iron heating units, and was least with direct radiators.

3. The rate of cooling of a room is greatly influenced by the heat capacity of the materials in the walls, floor, and ceiling. The evaluation of this effect is very difficult and uncertain.

4. The thermal energy of the steam in the unit at the time that the steam is turned off is comparatively small and has no effect on the rate of cooling of the room.

5. Under the venting conditions maintained on these tests, the air in the steam space was expelled and the heating surfaces attained full operating temperatures within 5 min after the steam was turned on. Under such conditions the rate at which the room heats is practically independent of the masses of the heating units, but is dependent on the capacities of the units and the effectiveness with which they emit heat to the air and the surroundings.

ACKNOWLEDGMENTS

The data presented in this paper were obtained in connection with an investigation conducted by the Engineering Experiment Station of the University of Illinois, of which M. L. Enger, Dean of the College of Engineering, is the Director. The investigation was conducted in the Department of Mechanical Engineering of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the Head. The results will ultimately comprise part of a bulletin of the Engineering Experiment Station.

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DISCUSSION

M. W. EHRLICH (WRITTEN): According to the objectives of the reported investigation, the aim was to get determinations equal to those prevailing in usual winter room heating, operating under practical service conditions.

It is submitted that apparently this mark has not been reached, because venting was at a constant rate regardless of the content or weight of the heating units; the temperature range was a variable, and the heaters were not of the same size.

The report and the conclusions seem all right in that they explicitly state that results apply only to the room in which tests were made under the specific conditions described. From the technical angle, therefore, this is an extensive piece of work and a commendable one, with considerable basic information. But so far as the practical value goes, it is unfortunate that some comments interspersed in the report, tend to distract from it. For the lay reader it might almost be called a bias in emphasizing the direct radiator,—for he has been subjected to commercial propaganda with the implication that only cast-iron is correct for heating,—whereas a wide experience factor, dating from the pioneering in convectors and the progress of this form of heating, shows that a copper convector is better than any radiator of equal height.

Take the data presented in the paper, with practical heating requirements in view, and they will confirm the advantages of the convector. Construct a composite curve and the result should take the form of Fig. A, shown here, with the values expressed in terms of equivalent temperature. It is self evident that during the first 30 min of time, the room condition was relatively best with the copper convector. In practice it would only be necessary to repeat or maintain this same cycle of heating and cooling to continue that desirable room condition, and do so with less steam consumption.

Being interested in non-ferrous convectors, particularly the fin-and-tube type of copper, further comment will compare the direct radiator with the copper convector.

Consider the facts, both as to operating characteristics shown, and the physical features of Test Units 1 and 5, as given:—

	COPPER CONVECTOR (UNIT 5)	DIRECT RADIATOR (UNIT 1)
Sq Ft, EDR	25.0	28.1
Height, In.	29	26
Relative weight	1	2.1
Relative volume	1	6.25
Temp. Range at 60 In.	14	11
Heating time, Min	24	82
Cooling time, Min	30	55
Ratio, Cooling Heating	1.25	0.68

It is seen from the accompanying tabulation that heaters are of slightly different size, therefore more steam was supplied to the direct radiator than to the convector; the convector is of greater height, although copper convectors are lower in usual practice, and their effectiveness depends upon height, with results being better as the height is decreased; there is quite a disparity in the internal volumes and so different venting time would apply in practice; the test range was intended for an 8-deg difference taken at the 30-in. level, but the copper convector had the highest dry-bulb range at the 60-in. level, which is still the height more nearly typical for the usual location of control devices in room heating practice.

With all this, the radiator took about 82 min to give the test room its temperature range while heating up, yet lost it again in but 68 per cent of that time; while the copper convector did its heating work in about 24 min, and took 125 per cent of that time to lose it.

In other words, the room equipped with a direct radiator was very slow in warming up, but it was somewhat faster in cooling down as compared with the

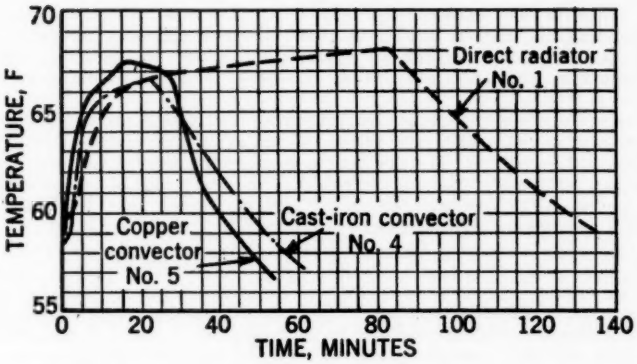

FIG. A. CHARACTERISTIC CURVES OF RADIATOR AND CONVECTORS (REPLOTTED FROM FIGS. 5, 6, 7 "b" AND "c")

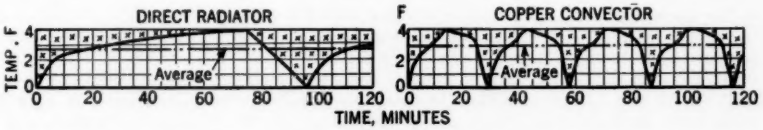

FIG. B. TEMPERATURE VARIATIONS WITHIN 4-DEG RANGE OVER A PERIOD OF TWO HOURS

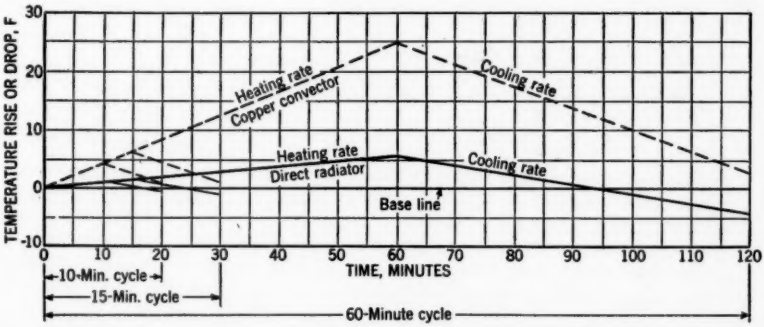

FIG. C. COMPARATIVE HEATING AND COOLING RATES FOR DEFINITE PERIODS OF OPERATION

time it took to fulfill the room heating requirement. Similarly the room fitted with a copper convector was warmed quite quickly but it was a little slower in cooling down as compared with its heating up time. These comparisons are for each unit with itself, and denote the relative behavior throughout the individual operating period, much as would take place in an installation using one or the other kind of heater.

This behavior shows two marked characteristics which plainly indentify the different forms of heating units, their mode of heat dissipation and distribution, their internal volumes and their weights. If there were to be a decision made as to choice for room heating purposes, surely that unit which warms the room quickly and responds with undue lag would be necessarily favored. And the copper convector more nearly meets this requirement and situation.

For examining the problem further, a smaller equivalent temperature range may be taken, such as would more nearly prevail in actual winter service operation. Retracing the curves of Fig. A, and extending them to cover a two-hour period, gives a result much like that in Fig. B, shown here. These fluctuations indicate a relatively lower average temperature with the radiator in the room; and a relatively higher average temperature with the convector in the room. If the marked spaces above the curves are said to be areas of *cold*, then there is a slightly greater total of cold with the radiator than with the convector.

Another way of looking at the same values is to analyze a hypothetical operation on a definite time basis. This has been done and transferred to Fig. C, by computation and extension of the temperatures on the same relative rates and terms as before. Of course it is purely an academic study, intended to accentuate the difference in the operating characteristics of a radiator and of a convector. It may serve as an illustration of the rates of heating and cooling in any theoretical operation for the several cycles shown. A cycle is to be understood as the time in which heating is done, and the same time allotted for cooling; thus a 10-min cycle would cover a total of 20 min. It is seen that in each case the convector again gives a better account of itself than the comparative performance of the radiator. Roughly, the convector shows about a 5-deg rise in 12 min, while the radiator takes 50 min to do about the same work.

These several comparisons should certainly go to confirm the results of other tests and the many practical observations over some period of time, that a direct radiator does take more than an hour to heat a given room, while a copper convector heats the same room in but a few minutes, with more comfort and better economy.

No. 1068

VENTILATION REQUIREMENTS

(Part 2)

By C. P. YAGLOU * (*MEMBER*) AND W. N. WITHERIDGE ** (*NON-MEMBER*)
BOSTON, MASS.

This is the second progress report of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the School of Public Health, Harvard University.

IN a previous paper [1] it was found that the number of persons occupying a room, or the air space per occupant, is a very important factor affecting the per capita outdoor air supply for the control of body odors. In a room with a net air space of 1410 cu ft, an outdoor air supply of 7 cfm per capita was found to be required when the room was occupied by three adult persons of average socio-economic status; 16 cfm per capita when occupied by seven persons and 25 cfm each with fourteen persons in the room.

Additional work presented in the present paper reveals two important factors responsible for this variation: (a) spontaneous disappearance of body odors with time and (b) changes in efficiency of ventilation and odor removal with changes in the amount of air supplied. A third factor suspected at first, namely, adsorption of odors on room surfaces, was found to be of little or no significance.

SPONTANEOUS DISAPPEARANCE OF ODORS IN AN UNVENTILATED ROOM

Disappearance of Body Odor

Heretofore body odor in the air of occupied rooms was regarded as a more or less stable entity, and the problem of odor control was thought to be mainly one of plain dilution with clean outside air. Evidence obtained during the past two years does not support this view, but indicates that body odors are very unstable, tending to disappear rapidly with time, much faster than most odors with which the ventilating engineer is confronted in public buildings.

All observations of body-odor disappearance rate were carried out in two

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** Instructor of Industrial Hygiene, Harvard School of Public Health.
[1] Ventilation Requirements, by C. P. Yaglou, E. C. Riley and D. I. Coggins, A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1937, by C. P. Yaglou.

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adjoining rooms, which were separately conditioned,[2] by shutting off all ventilation in the experimental room immediately after the subjects were dismissed, and smelling the strength of body odor at intervals by comparing it with the smell of air kept at threshold-odor intensity in the control room, as in the previous work.[3] The only significant leakage in the closed room was that occasioned in passing back and forth from the control room for determining

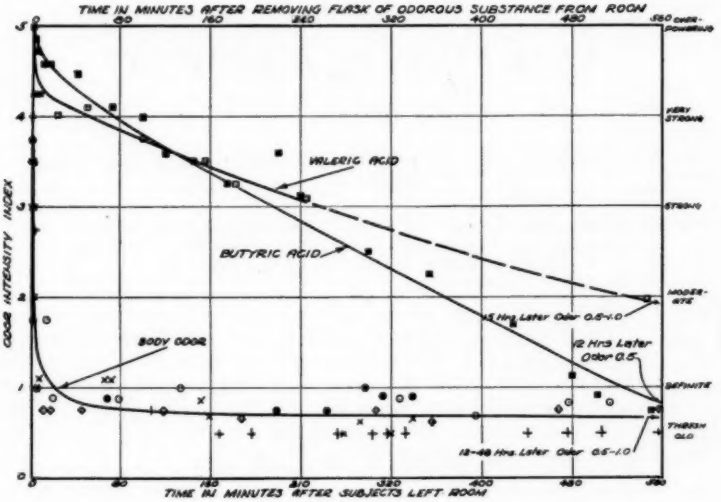

FIG. 1. DISAPPEARANCE OF BODY ODOR, VALERIC AND BUTYRIC ACID, IN A CLOSED ROOM

the odor intensity. By carefully opening and shutting the small door connecting the two rooms, this leakage was kept at a minimum.

In the control room the ventilation rate was maintained at 50 cfm per person throughout, as in the previous work, with not more than two persons in the room at a time. These were the judges who compared odor intensity. All experiments on body-odor disappearance rate were started about 12:30 p.m., after the subjects had been in the room for 3½ hours, from 9:00 a.m. to 12:30 p.m. Observations were continued to 6:00 p.m. or later. At night, the experimental room was closed tight and observations were resumed the following morning, with the normal rate of ventilation in the control room (50 cfm per occupant).

In the lower part of Fig. 1 are shown the results of four experiments carried out under representative conditions in winter and summer with respect to temperature, humidity, clothing, and other factors. It can be seen that in all

[2] Loc. Cit. See Note 1.
[3] Loc. Cit. See Note 1.

four instances the strength of body odor fell abruptly from an initial value between 3.5 and 4.0 to the allowable intensity of 2 in less than 4 min after the subjects left the room. Twenty minutes later the odor index was about 1. From then on, the rate of disappearance became very slow, and finally equilibrium was reached at an intensity between 0.5 and 1.0. By this time the odor lost much of its characteristic *human smell* and after 48 hours it was difficult to identify it as such in the presence of other weak odors coming from other sources in the room. A certain minimum amount of ventilation was always necessary to clear out this residual odor, which persisted indefinitely with the room kept tightly closed.

The explanation for the abrupt disappearance rate of body odor is not quite clear. Oxidation would seem to be a most plausible cause. It is possible that some constituents of the body-odor complex are extremely unstable, leaving behind other less odorous but relatively stable constituents which can be smelled for days. The loss of moisture from the organic material to the atmosphere may also have something to do with the loss of odor. Schroeder [4] recently presented some evidence of his own and others, indicating a chemical change in organic matter of expired air after washing with sulphuric acid. The alteration was attributed to dehydration, as the sulphuric acid did not remove an appreciable quantity of organic material. On the other hand, liquid condensed in the present experiments by cooling odorous air to about 35 F was odorless, clear and neutral in reaction. This is not necessarily contradictory with the work of Gant and Shaw,[5] who froze out odors from the air with dry ice and presumably succeeded in determining their relative strength several days later by the use of an osmoscope. The odors Gant and Shaw worked with were mainly those from tobacco smoke, foods and various kinds of liquors, whereas those in the experiments described in this paper were body odors alone.

The influence of per capita air space on ventilation requirements from the standpoint of body odor would, therefore, seem to be explainable, almost entirely, by the rapid disappearance of odor. The time element is so small that it is difficult to draw a satisfactory odor balance from the odor intensity in the room, number of occupants, air supply, and room dimensions. Even an error of half a minute or more in estimating the time taken by the ventilating current to pass over the occupied zone might make a great difference in the loss of odor and odor balance.

Disappearance of Other Odors

That the rapid loss of odor is due to a characteristic of body odor itself rather than to diffusion by leakage was ascertained by studying the disappearance rate of other stronger odors, such as valeric and butyric acid, tobacco smoke, and ozone, all capable of completely masking body odor. In the upper half of Fig. 1 are shown results with valeric and butyric acid, obtained by exposing in the center of the experimental room a small flask of the odorous substance, with the stopper removed, until the odor intensity rose to 5. With

[4] Elimination of Organic Products in Air Conditioning, by W. K. Schroeder, *Refrigerating Engineering*, 29, 294, Dec. 1933.
[5] Odor Concentration in Air-Conditioned Structures, by V. A. Gant and H. D. Shaw, *Industrial and Engineering Chemistry*, 9, 16, Jan. 15, 1937.

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valeric acid it took 25 min, and with butyric acid 30 min to raise the odor strength from 0 to 5. The flask was then removed and observations of odor intensity made at intervals, as in the case of body odors. By comparison with the body-odor curve in the lower half of Fig. 1, the rates of disappearance of valeric and butyric acid are much too slow, lending little support to the assumption often made that body odors are mainly composed of these two substances.

The disappearance characteristics of tobacco smoke are shown in Fig. 2. Curves *A* and *B* were obtained by having subjects smoke one or two cigarettes each inside the experimental room, with no ventilation at all. At the end of the smoking period, after the smokers left and the ash trays were removed, the odor intensity was determined at intervals as shown in previous tests.

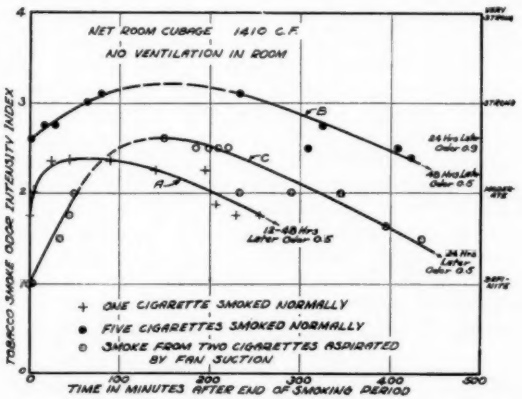

FIG. 2. DISAPPEARANCE OF TOBACCO SMOKE IN A CLOSED ROOM

Plot *C* (Fig. 2) was obtained with tobacco smoke from two cigarettes produced by fan suction, in order to check the characteristics of the other two curves. After the smoking period the fan and cigarette butts were removed from the room. In all three instances there was a bluish haze at first, particularly when smoke was produced by fan suction, gradually thinning out and changing to whitish color. This suggests flocculation to larger and larger particles, and eventual settling out of the particles.

Unlike body odor, the odor of tobacco smoke (Fig. 2) not only remained longer in the room, but its intensity increased in a peculiar manner during the first three hours following the smoking period. From then on, the intensity began to decrease gradually, finally reaching a stable level of 0.5 after 17 to 48 hours, according to the number of cigarettes smoked.

The rise in odor intensity after smoking seems to agree with common observation that the odor of stale tobacco smoke is more offensive than the odor of fresh smoke. Alterations in physical characteristics of the smoke in passing through the respiratory tract are probably not responsible for this, as shown

by curve *C*. Presumably, it would be advantageous to make smoking rooms as small as is practicable and ventilate them fast. The chemical and physical properties of cigarette smoke were recently reviewed by Bradford et al.[6]

Even ozone, which readily decomposes to O_2 and O, appears to be relatively more stable than body odor, as shown in Fig. 3. It is, however, the only one of the odoriferous substances studied by us that eventually disappeared completely without leaving a trace of residual odor of its own. Ozone for the tests was produced by corona discharge between concentric cylindrical electrodes, 5/8 in. apart, kept at a potential difference of 15,000 volts, a-c, 60 cycles. The outer cylinder was glass, lined with metallic foil on the outside. The apparatus was essentially a Cottrell precipitator as modified by Drinker,[7] for small-scale quantitative determination of dusts in air.

The decomposition rate of ozone on a volumetric basis was studied by Ewell [8,9] in a large closed box and in cold-storage rooms. The general char-

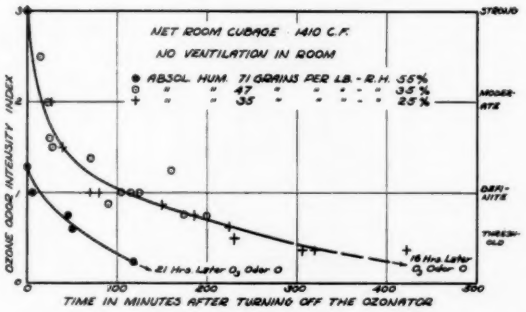
NET ROOM CUBAGE 1410 C.F.
NO VENTILATION IN ROOM
ABSOL. HUM. 71 GRAINS PER LB. - R.H. 55%
" " 47 " " " 35%
" " 35 " " " 25%
STRONG
MODERATE
DISTINCT
THRESHOLD
OZONE ODOR INTENSITY INDEX
21 Hrs. Later O3 Odor 0
16 Hrs. Later O3 Odor 0
TIME IN MINUTES AFTER TURNING OFF THE OZONATOR

FIG. 3. DISAPPEARANCE OF OZONE IN A CLOSED ROOM

acteristics of Ewell's curves are quite similar to those obtained in the tests described in this paper.

CHANGES IN VENTILATION EFFICIENCY

Aside from body-odor disappearance rate, the amount of air supply and the number and arrangement of occupants in a room with reference to the location of supply and exhaust openings affect also the efficiency of odor removal. With supply openings near the ceiling located centrally over the occupied zone, as in this system, and a constant air supply per person, an increase in the number of occupants would have two opposing effects on ventilation efficiency. On one hand, it would tend to increase efficiency by spreading out the occupied

[6] Nature of Cigaret Smoke, by J. A. Bradford, E. S. Harlow, W. R. Harlan and H. R. Hanmer, *Industrial and Engineering Chemistry, 29*, 45, Jan. 1937.
[7] Alternating Current Precipitator for Sanitary Air Analysis, by Philip Drinker, *Jour. Industrial Hygiene, 14*, 364, 1932.
[8] The Disappearance of Ozone in Cold-Storage Rooms, by A. W. Ewell, *Refrigerating Engineering, 26*, Dec. 1930.
[9] The Decomposition of Ozone, by A. W. Ewell, *Ibid., 29*, Aug. 1933.

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zone; on the other hand, the greater volume of air needed would allow less time for diffusion over the occupied area, the air passing quickly to the exhaust without removing a full share of odor, CO_2, and moisture.

The problem was studied by determining the relative efficiency of the ventilation system at various rates of air supply and with 3, 7, and 14 persons in the room.

Efficiency of ventilation for the purpose of this study was taken as the ratio of *effective* air supply to the total air supply. *Effective* air supply is that passing over the occupied zone, and capable of removing the products of respiration and transpiration. It was determined from measurements of carbon dioxide in representative areas, in the breathing zone between the seats of occupants. Most of the experiments were carried out with seven adult persons in the room

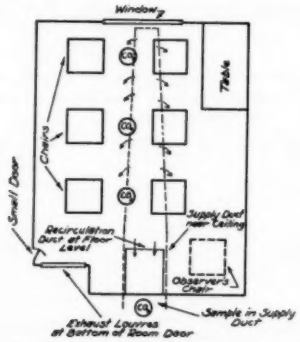

FIG. 4. PLAN OF EXPERIMENTAL ROOM SHOWING LOCATION OF SUBJECTS AND SAMPLING STATIONS

(see Fig. 4), the observer himself serving as subject. All experiments lasted 3½ hours, and the air samples were collected near the end of each test. Carbon dioxide analysis was made by means of a 20 cc modified Haldane apparatus for CO_2 only. The relative ventilation efficiency was computed from the following formula:

$$\text{Ventilation Efficiency}^{10} = \frac{0.01}{\dfrac{(CO_2)_r - (CO_2)_s}{\text{Cfm per occupant}}}$$

where

0.01 = cubic feet of CO_2 given off per person per minute.
$(CO_2)_r$ = average CO_2 of air in occupied zone, per cent by volume.
$(CO_2)_s$ = CO_2 in the air supply, per cent by volume.
Cfm per occupant = air supply per occupant in cubic feet per minute measured by means of standardized orifice meters.

It can be seen in Fig. 5 that with airflows under 5 cfm per person, ventilation efficiency was practically 100 per cent, decreasing progressively as the air supply increased. With 30 cfm per person, the efficiency was about 75 per cent.

[10] To obtain ventilation efficiency in per cent the ratio should be multiplied by 10,000, as Professor West suggests in the discussion at the end of this paper.

From the standpoint of efficiency, it would seem that rate of air supply is more significant than number of occupants in a room, but the data with three and fourteen persons are too few to draw a conclusion.

The important thing to emphasize is the advantage of large rooms over small ones, as they act like reservoirs, allowing body odor to disappear, with a minimum air supply and maximum ventilation efficiency. Reducing the size of a room entails an increase of ventilation rate and a simultaneous decrease of ventilation efficiency.

ADSORPTION OF BODY ODORS ON ROOM SURFACES

Adsorption is generally regarded as a physical, rather than a chemical phenomenon consisting in condensation of gas molecules on the surfaces of solid

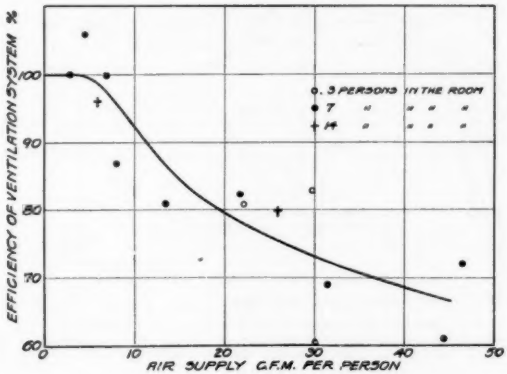

FIG. 5. EFFICIENCY OF VENTILATION SYSTEM OF EXPERIMENTAL ROOM AT VARIOUS AIRFLOWS

bodies with which they come in contact. The amount of gas adsorbed depends upon its pressure, the chemical natures and potentials of the adsorbents and adsorbate and the physical state of the adsorbing surface. For a great number of systems, adsorption is believed to be complete when the adsorbing surface is completely covered with a unimolecular layer of the adsorbent.[11] Normally, equilibrium is quickly reached when the rate of evaporation just balances the condensation rate. Exceptions occur when diffusion is slow, or when there is chemical interaction between the adsorbent and adsorbate. The length of time over which the condensed molecules will remain on the surface depends on the strength of the attractive forces between the surface and adsorbed molecules, and the temperature.

Under the conditions of the present experiments, adsorption of body odors on surfaces of walls, ceiling and floor did not seem to be a factor affecting odor intensity and ventilation requirements per occupant, when equilibrium

[11] International Critical Tables, McGraw-Hill Book Company, New York, *3*, 250, 1926.

conditions prevailed. This conclusion was arrived at by increasing the amount of adsorbing surface in the room and redetermining the equilibrium time, equilibrium intensity, and rate of disappearance of body odor under the new conditions. All other factors were kept approximately the same in each series of tests.

Plot *A* in Fig. 6 shows the normal variation of odor intensity in a control test with the normal amount of adsorbing surface of walls, ceiling, and floor aggregating 780 sq ft approximately. Odor equilibrium was reached about 50 min after the subjects entered the room, and the equilibrium intensity index averaged nearly 3.5. The disappearance rate in this control test is that already shown in Fig. 1 by similar points.

Plot *B* was obtained under identical initial conditions as plot *A*, except

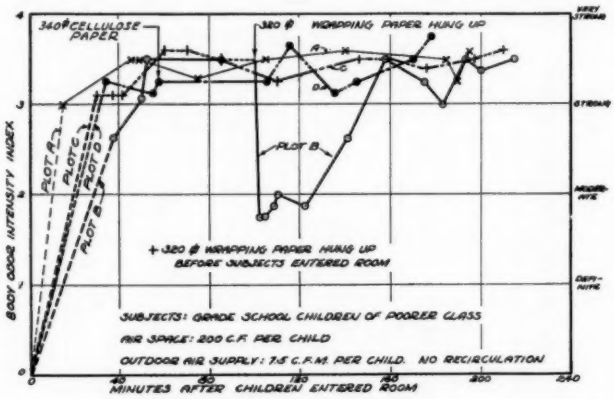

FIG. 6. EQUILIBRIUM TIME AND INTENSITY OF BODY ODOR WITH AND WITHOUT ADDITIONAL ADSORBING SURFACE IN THE ROOM

that near the middle of the test the adsorbing surface in the room was increased by 320 sq ft by hanging up sheets of semi-gloss brown wrapping paper vertically over the occupied zone and parallel to the air flow. The lower edges of the sheets hung down to shoulder height of the subjects.

The sudden decrease of odor intensity immediately after hanging up the sheets (Fig. 6) is probably due to a strong static charge acquired by the paper when it was unrolled freshly from a big roll just before hanging it up. The effect was transient, and the odor intensity soon rose to its former level, as can be seen in Fig. 6.

In plot *C* the same paper sheets were hung up in the room 16 hours before the subjects entered, with no apparent effect throughout the test period.

In plot *D* sheets of a spongy cellulose paper used in air filters were employed instead of wrapping paper, with negative results again.

Two experiments with an airflow of 30 cfm per person, instead of 7.5, likewise yielded negative results. The rate of disappearance of body odor at the end of the tests was substantially the same as in Fig. 1.

SUMMARY

Body odors are not stable but tend to disappear rapidly in a closed room within a few minutes after the occupants leave. A weak residual odor persists for days unless the room is ventilated.

The unusually rapid disappearance rate is a characteristic of body odor itself, differing greatly from disappearance characteristics of other odors, such as butyric or valeric acid, tobacco smoke, and ozone, all capable of completely masking body odor.

Unlike body odor, the odor of tobacco smoke not only remains longer in the air but its intensity increases during the first three hours following smoking.

Air space per occupant in a room not only affects the disappearance rate of body odors and hence ventilation requirements, but also the efficiency of ventilation systems. Large rooms have an advantage over small ones, as they act like reservoirs, allowing body odors to disappear with a minimum outdoor air supply and maximum ventilation efficiency. A small room would require a greater air supply per occupant for the control of body odor. An increased air supply entails a loss of ventilation efficiency, as the air passes quickly to the exhaust without removing a full share of odors, heat, and moisture.

On the other hand, the problem of tobacco-smoke control would seem to consist in making rooms as small as is practicable and ventilating them fast.

DISCUSSION

DR. W. J. MCCONNELL[12] (WRITTEN): Ventilation requirements for the elimination of odors have long been a matter of guesswork. The desirability of removing the causes of odors is recognized, but scarcely ever obtained.

The authors of this paper have materially contributed to our knowledge of the complex subject of odors and have offered practicable measures of reducing odor concentration to the point where it is no longer noticeable, when complete removal is impossible.

The charts indicating the disappearance of certain odors are extremely instructive in the absence of scientific instruments for satisfactorily measuring odors.

The report is a valuable addition to the studies conducted by Schroeder, Gant, Shaw and others on this important subject.

J. J. AEBERLY (WRITTEN): There are several reasons why a person about to discuss a technical paper should have the privilege of dealing in anticipations. It is in this sense that I wish to consider the data presented by Professor Yaglou and Mr. Witheridge.

The study of the character and minimization of odors is one of the moot questions within the society. The striking fact brought out in this presentation is that the odors under consideration are not gases; at least they do not follow some of the physical laws relating to vapor pressure and diffusion as do butyric and valeric acids when found in a vaporized state. The authors appear a little reluctant to state definitely that the odors are due to either liquids or solids; yet we may imply from the statement made in the text that some credence should be given to the thought that the rapid decline in odor intensity may be due to dehydration. To my way of thinking this explanation, namely, dehydration, is a much more reasonable deduction than the idea that oxidation may be the major cause of the rapid odor decline.

This work presents some new thoughts in relation to odors and the Society may

[12] Assistant Medical Director, Metropolitan Life Insurance Co., New York, N. Y.

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do some useful research work to determine exactly the physical and chemical characteristics of the substances causing the odors. The new micro technique developed in the field of industrial hygiene makes this problem less intricate than it has appeared in the past. In general, the authors have done a constructive piece of work.

I would like to call attention to the statement made in the summary: "An increased air supply entails a loss of ventilation efficiency as the air passes quickly to the exhaust without removing the full share of odors, heat and moisture." This statement is based upon calculations using a formula which involves the action of a gas and in varying concentrations. These concentrations are 24 parts per 10,000 parts of air in the case of an air supply of 5 cfm, and seven to nine parts when 30 cfm is used.

In view of the fact that the substances under consideration are not gases and since these data appear to relate especially to a small room the authors should indicate that this statement should not be taken too generally in its application. It is evident that the ratio of the floor area occupied by the persons in the room in question to the total floor area is less than the ratio that obtains in an auditorium or other place of public assembly and it is this ratio that in all probability has materially influenced the result.

I am still of the opinion that the efficiency as defined and measured by the authors is more closely related to the design of ventilating systems with a view to perfect distribution than in any variation of air supply. The authors, no doubt, will agree with me that if we had a uniform downward flow throughout an auditorium the efficiency of the ventilating system would be constant as measured and defined by the authors regardless of the varying air supplies that are generally used in ventilating systems.

C. M. ASHLEY (WRITTEN): Professor Yaglou's and W. N. Witheridge's paper clears up the large question mark left by the results given in an earlier paper and rounds out very admirably the work done on this highly important subject.

One result of this investigation, for which I think we can be particularly grateful to the authors, is that it effectively explodes the myth of air change based on room volume as a measure of ventilation. I consider it a real step forward if the idea of air change on a room volume basis can be eliminated from our thinking.

It seems to me that the next logical step which should flow from this work is to attempt to set up a formula or method by which various grades of room odor intensity can be predicted for any given installation. It should thus be possible to set up the standards necessary to produce a Class A, B or C job from the standpoint of odor. Doubtless this work will be carried on through a committee of the Society, but such an attempt to set up an odor standard may reveal further gaps in our knowledge which have not as yet been covered by Professor Yaglou's and Mr. Witheridge's work. This work should, I believe, act as a guide in any future work which they carry on.

I have been somewhat concerned about the amount of infiltration which might be permitted through the small door used by the observer and I have wondered if a refinement of technique could not be made by using only an opening sufficient in size to permit the observer's head to project through from one room to another. It seems to me fairly certain that some of the diminution of odor intensity must be due to infiltration of air, although the marked difference between the results for body odor and various other odor sources indicates the impossibility of assigning a large role to this cause.

It seems to me that the question of surface adsorption of odor deserves somewhat further study if an unqualified answer is to be given, although I am not sure that the practical importance of this justifies more effort. Is it not possible that there is

an adsorption to an equilibrium condition beyond which the effect of additional surface is not apparent? This suggestion would necessarily assume that the rate of decay of odor on a surface was not appreciable. This question undoubtedly must be answered by further study of the character of body odor.

PROF. PERRY WEST (WRITTEN): This is unquestionably a most interesting paper and is no doubt pointing the way to some very new and valuable conceptions of ventilation requirements.

The general conclusions that a larger room volume per occupant will permit a smaller per capita air supply from out-of-doors for the control of body odors, while a smaller room volume per occupant with greater per capita out-of-door air supply is more efficient for the control of odors from tobacco smoke, etc., are quite interesting and useful.

I feel, however, and the authors will no doubt agree, that enough work has not been done on these subjects to properly define the exact effect that the room volume has upon either of these conditions.

In the previous paper referred to, where the room volume was 1410 cu ft, the cfm from out-of-doors required for the control of body odors would work out as shown in Table A.

The variation here is quite wide and somewhat out of keeping with what we have come to consider good ventilation. For instance in a theater where the cubic contents might range from 200 to 300 cu ft per person the air supply from out-of-doors will range from 10 to 7 cfm per person, instead of from 16 to 11 as the above might indicate.

I note in connection with Fig. 4 that there is a window and a door in the test room employed in these experiments and there are no doubt other sources of air leakage, including the natural cracks and pores in the walls, floor and ceiling.

No mention is made of any account having been taken of the natural air movement

TABLE A

NO. OF OCCUPANTS	ROOM VOL. CU FT PER OCCUPANT	CFM REQUIRED FROM OUT-OF-DOORS PER OCCUPANT TO CONTROL BODY ODORS
3	470	7
7	201	16
14	100	25

TABLE B

NO. OF OCCUPANTS	ROOM VOL. CU FT PER OCCUPANT	NATURAL AIR CHANGE CFM PER OCCUPANT	ADDITIONAL CFM REQUIRED TO BE FORCED IN FROM OUT-OF-DOORS PER OCCUPANT TO CONTROL BODY ODORS	TOTAL CFM REQUIRED PER OCCUPANT FROM OUT-OF-DOORS
3	470	8	7	15
7	201	3.3	16	19.3
14	100	1.7	25	26.7

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into and out of the room upon which the tests for determining the effect of room space upon air supply was based.

In a small room of this character where the volume per occupant is relatively small as compared with office space, anything like a normal air leakage would, of course, have considerable weight upon the results.

If, for instance, we should figure one air change per hour the results would be as shown in Table B.

If such a condition should obtain, then in the first case, as shown in Table A, the ventilation requirements would vary 360 per cent per capita and the ventilation efficiency would vary from 100 per cent down to 28 per cent, whereas from Table B the ventilation requirements would only vary 175 per cent and the efficiency from 100 to 50 per cent.

I note in this connection from Fig. 5, that the ventilation efficiency varies from about 97 per cent for 7 cfm per person to about 77 per cent for 25 cfm per person, which is a difference of 20 per cent, whereas the difference in efficiency indicated by the requirements of 7 cfm and 25 cfm per person is 72 per cent.

This difference might be explained, as the authors suggest, by the rapid spontaneous disappearance of body odors and the fact that this rate of disappearance is supposed to be a function of the room volume per capita, but I fail to find much data to support this, since all of the tests upon which curves in Fig. 1 are based were apparently made with the same number of occupants in the room.

Under the heading of "Changes in Ventilation Efficiency" I do not understand the formula—

$$\text{Ventilation Efficiency} = \frac{0.01}{\frac{(CO_2)r - (CO_2)s}{\text{cfm per occupant}}}$$

As an example take the following:

cfm per occupant = 5
$(CO_2)r$ = 0.2 per cent
$(CO_2)s$ = 0.0 per cent

Then—

$$\text{Ventilation Efficiency} = \frac{0.01}{\frac{0.2 - 0.0}{\frac{5}{1}}} = 0.01$$

Whereas the authors say that the efficiency for 5 cfm = 100 per cent.
It would seem, therefore, that this formula should read—

$$\text{Ventilating Efficiency in per cent} = \frac{0.01}{\frac{(CO_2)r - (CO_2)s}{\text{cfm per occupant}}} \times 10{,}000$$

Take as another example —

cfm per occupant = 5
$(CO_2)r$ = 0.24 per cent
$(CO_2)s$ = 0.04 per cent

Then —

$$\text{Ventilating Efficiency in per cent} = \frac{0.01}{\frac{0.24 - 0.04}{\frac{5}{1}}} = 0.01$$

which again must be multiplied by 10,000 to give results in per cent.
Would it not be simpler to write the formula —

$$\text{Ventilation Efficiency in per cent} = \frac{0.01 \times 10{,}000}{\dfrac{(CO_2)r - (CO_2)s}{\text{cfm per occupant}}}$$

$$= \frac{100}{\dfrac{(CO_2)r - (CO_2)s}{\text{cfm per occupant}}}$$

Some further enlightenment on the above points will be of greatest interest to me and I believe to some of our other members.

PROF. C. P. YAGLOU: Most of Mr. Aeberly's comments on the identity of the body-odor complex are, I believe, too speculatory to be discussed at this time with the limited knowledge that we have. It would, however, be more reasonable to assume that body odors may be due to gaseous, liquid, and solid matter, than to any one of the three at the exclusion of the other two.

I also cannot agree with Mr. Aeberly's belief that the ventilation efficiency, as defined in the text, should remain constant regardless of changes in the air supply. If we make simultaneous observations of absolute humidity or carbon dioxide in the occupied zone and in the exhaust of a tight room supplied with any given distribution system, we shall find that our measurements under equilibrium conditions are invariably higher in the occupied zone than in the exhaust. The greater the air flow, the greater the difference between the two sets of measurements. If the room is cooled in warm weather, the temperature, too, will often be lower in the exhaust than in the occupied zone, except when the walls are considerably warmer than the air. The main reason appears to be that air diffusion is not perfect, except under very low air flows, and the greater the air supply to a given room, the poorer the diffusion, because the fresh air has less and less opportunity to mix with the room air. The data in Fig. 5 speak for themselves. They apply to the experimental rooms under consideration and to the given distribution system. In any other system, the efficiency may vary over wide limits. In a series of tests carried out in Massachusetts public schools by Mr. McPherson, Dr. Keenan, Professor Holt, and myself, ventilation efficiencies varied from 20 to 90 per cent depending largely on the type of system, location of supply and exhaust registers, and quantity of air supplied.

Professor West, apparently, has missed an important point in the first report on ventilation requirements, where it is stated that all cracks and crevices in the experimental rooms were carefully sealed with plasticene and adhesive tape, and that the only significant leakage occasioned was that through the small connecting door when the *judges* passed from the control to the experimental room, once every hour, or so. This door closed tightly against heavy felt strips and the main room door was sealed tight and kept locked. The walls and ceilings of the rooms were painted with two coats of glossy enamel and the concrete floor was covered with linoleum cemented in place.

Under such conditions, Professor West will, I believe, agree with me that his Table B does not hold. Natural air leakage that may occur in an ordinary room should be subtracted (instead of added) from the air requirement for the control of body odor in order to estimate the probable outdoor air supply by fan. Moreover, when a spray washer is used and the spray water is changed daily, the ventilation requirements are considerably lower than those in Table A, as shown in the first report of the series. This has a direct bearing upon Professor West's reference to lower practical requirements of theaters.

Professor West is perfectly correct in assuming that the right hand side of the equation for ventilation efficiency should be multiplied by 10,000 in order to express the result in per cent.

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In closing, I should like to make clear that the main purpose of this laboratory study was to ascertain the factors affecting odor intensity in occupied rooms under controlled conditions, and to study suitable methods of odor control. Insofar as I know, no claim has ever been made that the data can be applied directly to practice, although there is considerable evidence to suggest that better results can be secured by the application of such fundamental data than by the use of rule-of-thumb methods in arriving at the probable outdoor air supply for body-odor control in various buildings. Field observations that have been in progress for some time will undoubtedly be of value in a final coordination of basic principles with practical facts.

No. 1069

HEATING REQUIREMENTS OF AN OFFICE BUILDING AS INFLUENCED BY THE STACK EFFECT

By F. C. HOUGHTEN * (*MEMBER*), AND CARL GUTBERLET ** (*NON-MEMBER*)
PITTSBURGH, PA.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

REVISION of the heat transfer constants, based on laboratory tests at the University of Minnesota, the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and elsewhere, and published in THE A.S.H.V.E. GUIDE 1933, forcibly called to the attention of designing engineers the need for a better understanding of other factors entering into the estimation of heating requirements of buildings. These other factors include the heat capacity of the building, the effect of solar radiation, wind velocity and direction, and the co-incidental effect of low temperature and high wind velocity in effecting a maximum rate of heat loss. The stack effect of tall buildings, together with the probable variation in wind velocity and direction near the base and upper stories of such a building in relation to that reported by the Weather Bureau, are also important factors. Other factors involved include the characteristic operation or usage of the building, the resistance to air circulation through different parts of the building, particularly to air flow upwards from floor to floor in the case of tall buildings, and heat sources within the building itself, such as mechanical equipment, lighting, and the human load.

An intensive study of these factors was initiated with the appointment of a Technical Advisory Committee on Heat Requirements of Buildings in the Fall of 1933 under the Chairmanship of D. S. Boyden. This work has been carried on under the following continuing committee for 1937: O. W. Armspach, *Chairman*, P. D. Close, W. H. Driscoll, H. M. Hart, V. W. Hunter, H. H. Mather, E. C. Rack, F. B. Rowley, R. J. J. Tennant and J. H. Walker.

One important phase in the work of this committee was the collection of a great mass of data during the heating season of 1933-1934, in eight rooms located on three different floors and on three different exposures in the Grant Building in Pittsburgh. Analysis of these data and the publication of

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the results have been a laborious procedure, requiring a considerable portion of the time of part of the Research Laboratory's staff in succeeding years. In 1934 a report [1] covering the relation between the wind velocity in close proximity to the building to that reported by the Weather Bureau was presented to the Society. In 1936 a more comprehensive report [2] of the relation between the electrical heat input required by the different rooms studied to the outside weather conditions and the location of the room in the building was published. In these two reports, the general question of aeration of the building, including infiltration and exfiltration as it is affected by the elevation of rooms within the building, or the stack effect, was given brief consideration. The more complete analysis of the data bearing on these phases of the subjects is included in this report.

FIG. 1. THE GRANT BUILDING SHOWING LOCATIONS OF SIX OF THE EIGHT ROOMS USED IN THE STUDY

A photograph of the building, showing the location of the several rooms in which the study was made, is presented in Fig. 1. The floor plan, Fig. 2, shows the orientation of the building. Table 1 gives other characteristics concerning the several rooms studied, including the estimated heating requirement based upon transmission and infiltration losses, as determined in accordance with current GUIDE methods. The table also includes the relative heat requirements of the several rooms.

In order to obtain a better understanding of the infiltration for the different rooms under actual test conditions, a number of tests were made of the rate of infiltration by the carbon dioxide method. The building is equipped with heavy casement rolled section steel sash windows. In the rooms studied the windows were of approximately the same size and tight fitting. Probably for this reason the infiltration observed for the different rooms showed little or no variation when related to the actual pressure drop through the window. Any consistent variation in such leakage for different rooms amounted to no

[1] Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw and Carl Gutberlet, A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 387.
[2] Heating Requirements of an Office Building as Affected by Weather Conditions, by F. C. Houghten and Carl Gutberlet, A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936.

more than the variation indicated by successive tests in the same room. The relation between pressure drop through the windows and the rate of infiltration is given in Fig. 3. All of the rooms studied were of approximately the same size and contained one window, with the exception of room 2706, which was a corner room of approximately the same size but with double outside wall exposure and containing two windows.

A more comprehensive discussion of the test procedure, method of calculating heat loss by transmission, the electrical heat input, and the heat gain by solar radiation are included in the earlier Laboratory reports.[3] A comparison of calculated, and measured rates of heat loss and heat input, by infiltration, transmission, solar radiation and electrical input are shown in Figs. 4, 5 and 6 for the eight rooms under various outside temperatures.

Values taken from the curves at 50 F temperature difference are given for comparison in Table 2. The calculated infiltration loss based upon THE GUIDE formula,

$$M_e = \sqrt{M^2 + 1.75a}$$

where:

M_e = equivalent wind velocity affecting pressure drop through the window.
M = wind velocity upon which infiltration would be determined if temperature difference were neglected, usually 15 mph.
a = distance of windows under consideration from mid-height of building; negative if above, positive if below.

is given by Curve F and Column D for each room.

The heat loss by infiltration based upon the observed pressure drops between the outside and inside is given by Curves G and Column C, when these values are positive. On the 27th floor, the stack effect was sufficient to oppose the effect of wind velocity in almost every observation made, with the result that in the average case a considerable exfiltration or negative infiltration resulted. Since for most considerations exfiltration does not result in negative heat loss or positive heat gain from the building, Curves G are not provided for rooms on the 27th floor, and heat loss values by infiltration are indicated as zero in Column C. The justification for this assumption will be discussed at greater length later in this paper.

The heat input by electric heaters used in maintaining room temperatures is given by Curves D, and the sum of the electrical input and 50 per cent solar radiation is given by Curves C. Since rooms 705 and 2703 facing northeast, received no appreciable solar radiation, Curves C and D are the same. Curves C and Column F represent the measured total heat input required to maintain the room at the predetermined temperature. Curves A and B for each room represent the estimated heat requirement by transmission and infiltration based upon THE GUIDE formula, and by transmission and infiltration calculated from the observed pressure drop, respectively.

The agreement between Curves C and B on the one hand, and between Curve C and Curve A on the other, for each room, indicates the degree of agreement between the actual heat input and the estimated heat requirement based on infiltration calculated by the two methods. These comparisons are given on a percentage basis in Columns G and H. It will be observed that considerably better agreement is had when the infiltration is based upon the observed pressure drop through the window. The average percentages by

[3] Loc. Cit. See Notes 1 and 2.

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floors given in Column G show fairly good agreement, or percentages near 100 for the 7th and 18th floors, but poor agreement for the 27th floor. The lower percentage of 89.8 for the 27th floor indicates that even when no heat loss

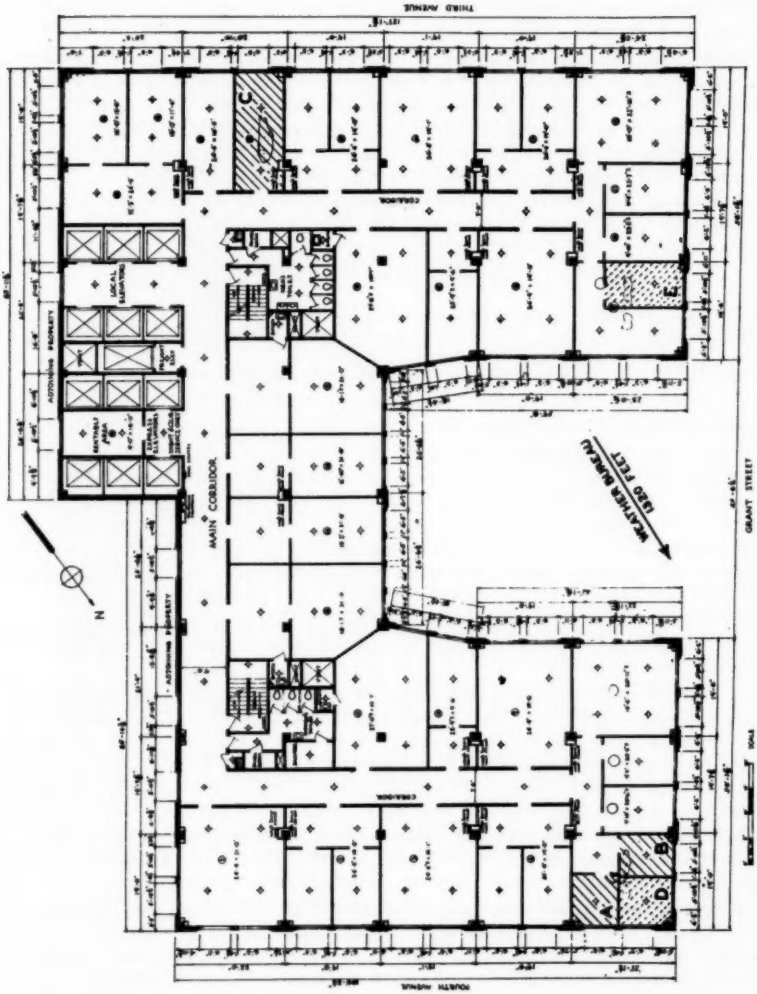

FIG. 2. PLAN OF 4TH TO 23RD FLOORS OF GRANT BUILDING, SHOWING ROOMS A, 705; B, 707; C, 726; AND 1826. ROOMS 2703, 2706, 2715 AND 2718 ARE SIMILARLY SITUATED TO A, D, E AND C RESPECTIVELY, BUT SET BACK FROM THE BUILDING

by infiltration is charged to the 27th floor, the calculated heat loss is measurably greater than the heat supplied. This suggests that the assumption that there is no negative heat loss by infiltration on the 27th floor where air

TABLE 1—ROOM CHARACTERISTICS

ROOM NO	EXPOSURE		WALL AREA SQ FT		TRANSMISSION EXPOSED WALL INCLUDING GLASS BTU/HR/DEG. F TEMP. DIFF	FACTOR TO CORRECT EXPOSURE OF ALL ROOMS TO 1828
	DIRECTION	FACING	MASONRY	WINDOW		
705	N.E.	4TH AVE	42.62	40.75	84.0	1.04
707	N.W.	GRANT ST.	51.26	38.24	83.8	1.05
726	S.W.	3RD AVE.	60.21	40.68	89.8	.96
1828	S.W.	3RD AVE	50.59	40.75	88.5	1.00
2703	N.E.	4TH AVE.	37.87	40.87	68.2	.96
2708	N.E. & N.W.	4TH AVE. GRANT ST.	123.14	79.66	139.5	.48
2715	N.W.	GRANT ST.	55.09	37.91	84.8	1.03
2718	S.W.	3RD AVE	61.81	40.62	70.4	.95

is blown out may be in error, or in other words, that there is actual heat gain by exfiltration for these rooms.

Due to the stack effect of the building, cold air flows in through every crack and crevice on the lower floors, passes upward through the building and outward in equal mass through cracks and crevices in the windows and walls of the upper floors. Every pound of outside air at its prevailing temperature entering the lower part of the building must be heated to 70 F, in order to maintain room temperature, and hence, the assumption with regard to this heat requirement cannot be much in error. On the other hand, air leaking into rooms in the upper part of the building from the floors below does not need to be heated up before leaving. It must, however, be cooled down to the prevailing outside temperature somewhere in its passage from the uniformly heated portion of the room to the uniform outside temperature some distance from the building. A real understanding of just where this cooling down process takes place would serve to give a better understanding of what actually takes place in the building.

In rooms in the upper part of the building there must be a resultant flow

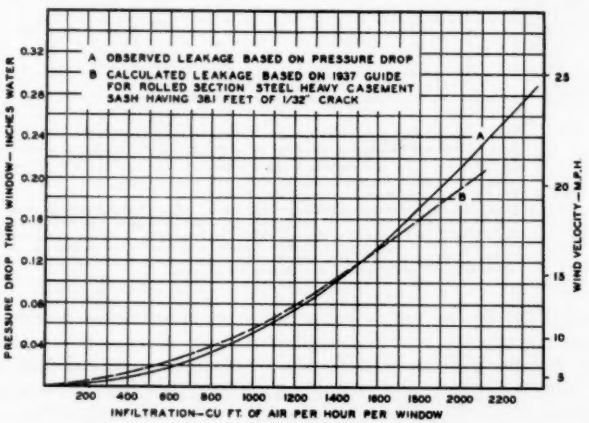

FIG. 3. RELATION BETWEEN INFILTRATION THROUGH THE WINDOWS OF THE GRANT BUILDING AND PRESSURE DROP

of air from the center portion of the room toward the exposed glass and walls, and more particularly toward the cracks and crevices in the same. This will naturally result in a somewhat higher temperature near the glass and walls where it finds its exit than would be the case if no air flowed outward. Hence, some of the heat flowing outward through the glass and walls by transmission

TABLE 2—RELATION BETWEEN HEAT SUPPLIED TO MAINTAIN TEMPERATURE AND ESTIMATED HEAT LOSSES FOR THE ROOMS STUDIED

A	B	C	D	E	F	G	H	I	J
ROOM NUMBER	CALCULATED TRANSMISSION BTU/HR./F. TEMP. DIFF.	CALCULATED INFILTRATION BASED ON P.D. B.T.U./HR./F. T.D.	CALCULATED INFILTRATION (GUIDE FORMULA) B.T.U./HR./F. T.D.	CALCULATED INFILTRATION BASED ON P.D. B.T.U./HR./F. T.D.	INPUT—ELECTRICAL PLUS 50% SOLAR RAD. B.T.U./HR./F. T.D.	ELEC. PLUS 50% SOLAR RAD. CAL. TRANS.+INF. COL. $\frac{F}{B+E}$ X 100 %	ELEC. PLUS 50% SOLAR RAD. CAL. TRANS.+INF. COL. $\frac{F}{B+D}$ X 100 %	ELEC. PLUS 50% SOLAR RAD. CAL. TRANS.+INF. COL. $\frac{F}{B+E}$ X 100 %	COL. I CORRECTED FOR SOLAR EFFECTIVENESS OF 50% RADIATION ON GRANT & FOURTH AVES.
705	84.0	32.0	42.0	32.0	88.0	91.7	83.0	91.7	98.7
707	83.8	34.0	42.0	34.0	107.5	110.1	101.8	110.1	107.1
726	89.8	34.2	42.0	34.2	108.5	102.8	95.4	102.8	102.8
1828	86.5	23.2	30.6	23.2	88.7	98.9	91.3	98.9	98.9
2703	89.2	0.0	18.0	-10.8	55.0	79.5	63.1	93.9	98.9
2708	139.5	0.0	36.0	-19.0	126.0	90.3	71.8	105.0	101.4
2715	84.8	0.0	18.0	- 8.8	83.3	97.7	78.4	112.4	104.8
2718	70.4	0.0	18.0	- 8.0	64.5	91.8	73.0	103.4	103.4
3RD AVE. 726 1828 2718						97.7	88.6	101.8	101.8
4TH AVE. 705 2703						85.8	73.0	92.8	97.8
GRANT 707 2715						103.9	89.1	111.2	108.0
7TH FLOOR						101.5	93.4	101.5	102.1
18TH FLOOR						98.9	91.3	98.9	98.9
27TH FLOOR						89.8	71.1	103.7	102.1
ALL						95.3	82.0	102.2	101.7

COL. B BASED ON THE GUIDE HEAT TRANSMISSION COEFFICIENTS AND 50° F TEMPERATURE DIFFERENCE.

COL. C BASED ON THE AVERAGE OBSERVED PRESSURE DIFFERENCE THROUGH THE WINDOW FOR ALL TESTS, THE OBSERVED RELATION BETWEEN THE PRESSURE DIFFERENCE AND AIR FLOW AND A 50° F TEMPERATURE DIFFERENCE FOR 7TH AND 18TH FLOORS AND NO HEAT LOSS BY INFILTRATION FOR THE 27TH FLOOR.

COL. D BASED ON INFILTRATION BY GUIDE FORMULA

COL. E BASED ON THE AVERAGE OBSERVED PRESSURE DIFFERENCE THROUGH THE WINDOW FOR ALL TESTS, THE OBSERVED RELATION BETWEEN THE PRESSURE DIFFERENCE AND AIR FLOW, AND A 50° F TEMPERATURE DIFFERENCE FOR 7TH AND 18TH FLOORS AND NEGATIVE HEAT LOSS FOR 27TH FLOOR BASED ON SAME PRESSURE AND INFILTRATION RELATIONSHIP BUT TEMPERATURE DIFFERENCE TAKEN FROM ROOM AIR TO GLASS.

will be provided by the falling temperature of the outward flowing air as it approaches the exposed wall and window. While strictly speaking it may not be agreed that this represents heat addition to the room by exfiltration, it must result in a decreased heat loss from the air within the room. A true evaluation of this effect would be given by a properly integrated temperature of the air streams leaving through the cracks. No observations were made of these temperatures, but if they are assumed to be the glass surface temperatures,

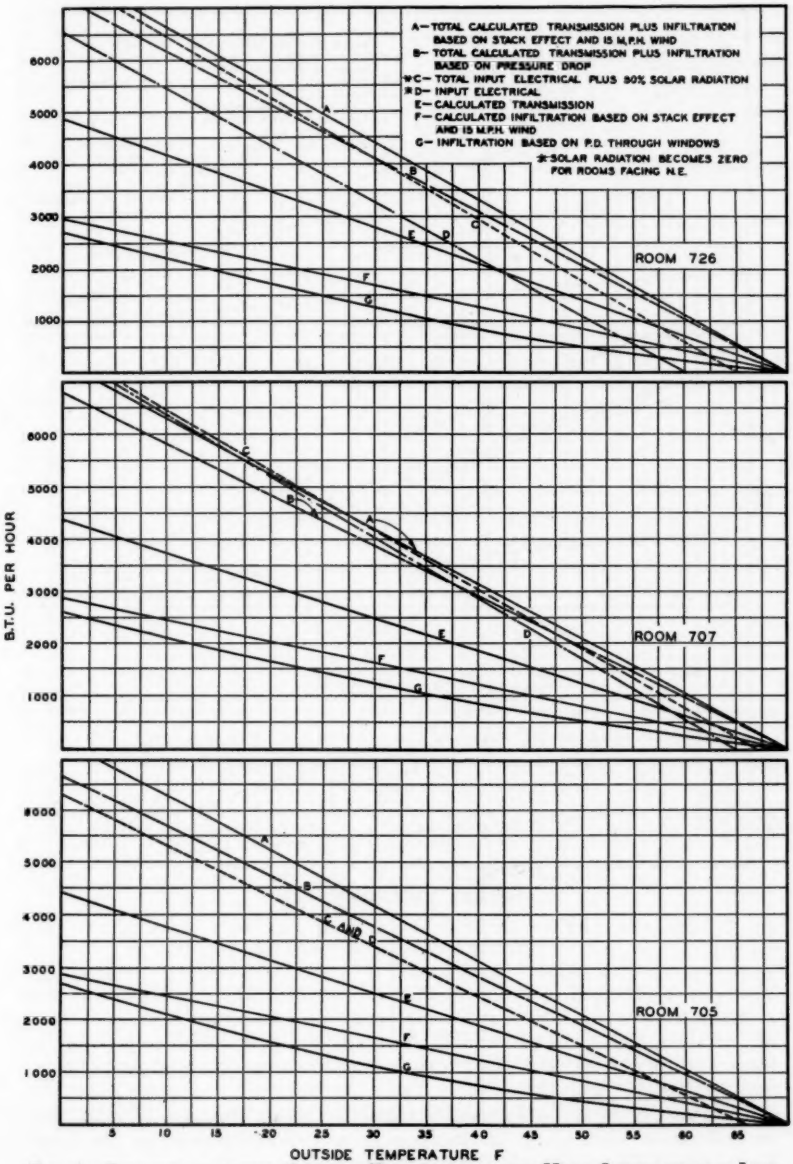
A—TOTAL CALCULATED TRANSMISSION PLUS INFILTRATION BASED ON STACK EFFECT AND 15 M.P.H. WIND
B—TOTAL CALCULATED TRANSMISSION PLUS INFILTRATION BASED ON PRESSURE DROP
*C—TOTAL INPUT ELECTRICAL PLUS 90% SOLAR RADIATION
*D—INPUT ELECTRICAL
E—CALCULATED TRANSMISSION
F—CALCULATED INFILTRATION BASED ON STACK EFFECT AND 15 M.P.H. WIND
G—INFILTRATION BASED ON P.D. THROUGH WINDOWS
*SOLAR RADIATION BECOMES ZERO FOR ROOMS FACING N.E.
ROOM 726
ROOM 707
ROOM 705
B.T.U. PER HOUR
OUTSIDE TEMPERATURE F
C AND D

FIG. 4. RELATION BETWEEN OUTSIDE TEMPERATURE AND HEAT INPUT TO AND LOSS FROM ROOMS ON THE 7TH FLOOR

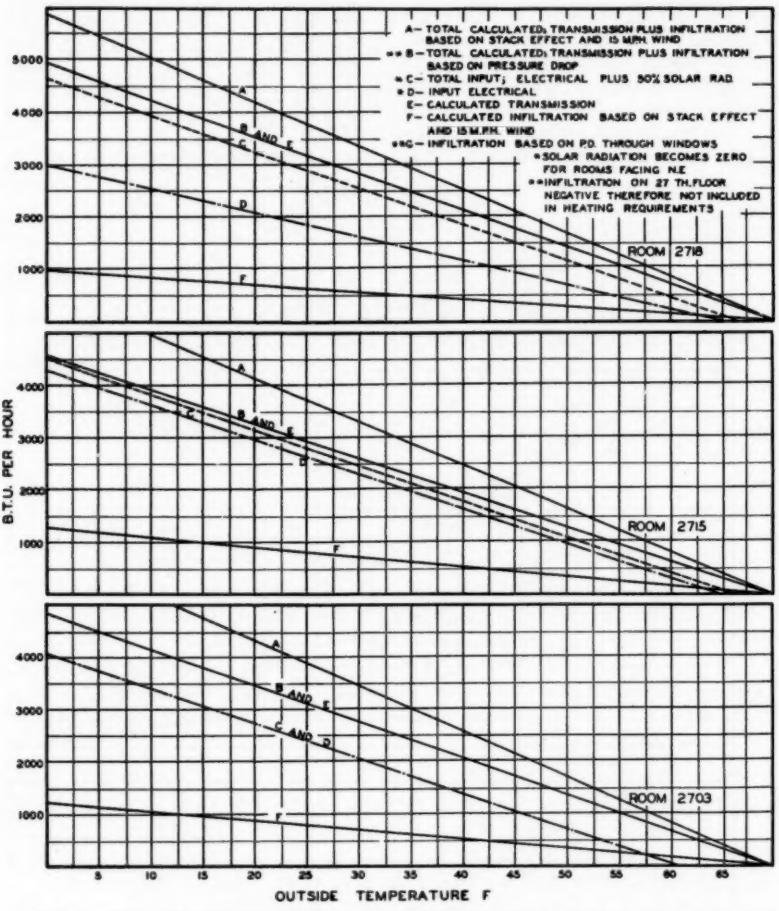
A— TOTAL CALCULATED; TRANSMISSION PLUS INFILTRATION BASED ON STACK EFFECT AND 15 M.P.H. WIND
**B— TOTAL CALCULATED; TRANSMISSION PLUS INFILTRATION BASED ON PRESSURE DROP
*C— TOTAL INPUT; ELECTRICAL PLUS 50% SOLAR RAD.
*D— INPUT ELECTRICAL
E— CALCULATED TRANSMISSION
F— CALCULATED INFILTRATION BASED ON STACK EFFECT AND 15 M.P.H. WIND
**G— INFILTRATION BASED ON P.D. THROUGH WINDOWS
*SOLAR RADIATION BECOMES ZERO FOR ROOMS FACING N.E.
**INFILTRATION ON 27 TH. FLOOR NEGATIVE THEREFORE NOT INCLUDED IN HEATING REQUIREMENTS
ROOM 2718
ROOM 2715
ROOM 2703
B.T.U. PER HOUR
OUTSIDE TEMPERATURE F

FIG. 5. RELATION BETWEEN OUTSIDE TEMPERATURE AND HEAT INPUT TO AND LOSS FROM ROOMS ON THE 27TH FLOOR

which were observed in the tests, then the negative heat loss by exfiltration given in Column E, Table 2, will pertain, giving an average balance of 103.7 per cent for all rooms on the 27th floor. Undoubtedly, the assumption that the air leaves the effective sphere of the room at glass temperature gives too high a value for heat gain.

Obviously, if there is a decrease in transmission loss from the 27th floor due to exfiltration, then, a similar increase in loss should pertain on the lower floors, where the air flows in the opposite direction.

A better comparison of the relation between the observed pressure drops

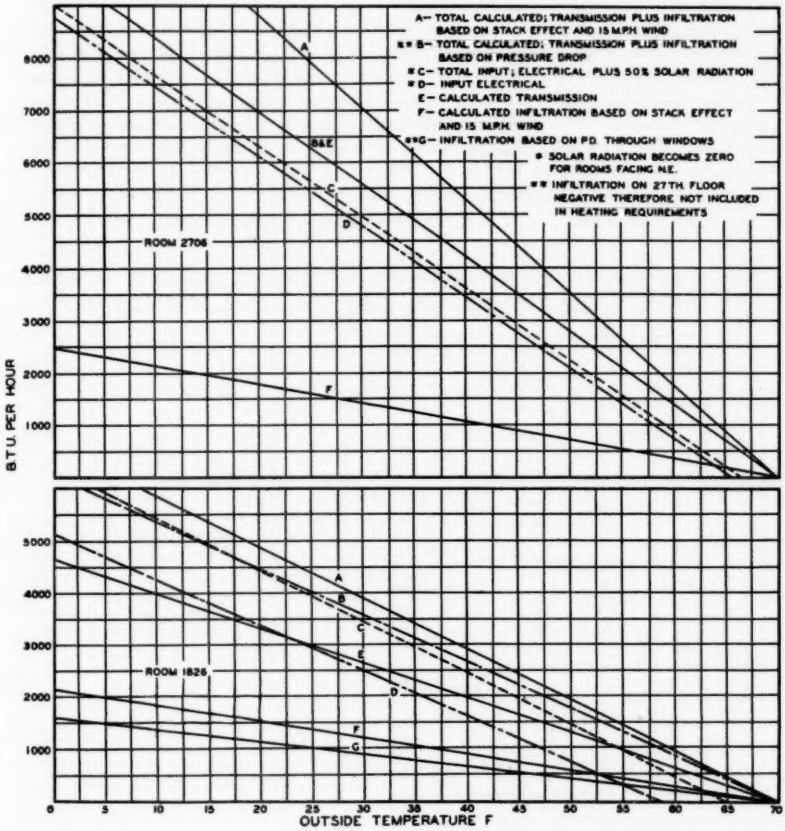
A— TOTAL CALCULATED; TRANSMISSION PLUS INFILTRATION BASED ON STACK EFFECT AND 15 M.P.H. WIND
*B— TOTAL CALCULATED; TRANSMISSION PLUS INFILTRATION BASED ON PRESSURE DROP
*C— TOTAL INPUT; ELECTRICAL PLUS 50% SOLAR RADIATION
*D— INPUT ELECTRICAL
E— CALCULATED TRANSMISSION
F— CALCULATED INFILTRATION BASED ON STACK EFFECT AND 15 M.P.H. WIND
**G— INFILTRATION BASED ON P.D. THROUGH WINDOWS
* SOLAR RADIATION BECOMES ZERO FOR ROOMS FACING N.E.
** INFILTRATION ON 27TH. FLOOR NEGATIVE THEREFORE NOT INCLUDED IN HEATING REQUIREMENTS
ROOM 2706
ROOM 1826
B.T.U. PER HOUR
OUTSIDE TEMPERATURE F

FIG. 6. RELATION BETWEEN OUTSIDE TEMPERATURE AND HEAT INPUT TO AND LOSS FROM ROOMS 1826 AND 726

through the windows and the theoretical stack effect, for various outside temperatures is shown in Fig. 7. THE GUIDE formula for pressure drop through windows at different elevations in a building, given previously, and based upon a 15 mph wind, is only meant to apply for a 70 F temperature drop, or for a zero deg outside temperature. These values are indicated for the three floors and a zero outside temperature. It should be noticed, however, that these values are for a 15 mph wind, while the average wind velocity resulting in the *measured* curves averages more nearly 7 mph. The *theoretical* values are for still air.

In further analysing the percentage of heat input based on the estimated heat loss, as given in Column I, it will be observed that room 707 and 2715,

facing northwest or Grant Street, show the highest percentages, or 110.1 and 112.4, respectively, while room 705 and 2703, facing the northeast or Fourth Avenue, give the lowest percentages, or 91.7 and 93.9, respectively. The reason for these discrepancies cannot be found easily. In fact, actually they are not larger than might be expected as the result of uncontrolled variables encountered in the problem. However, since they follow exposure direction rather than any other arrangement, it is safe to assume that the variation is not due to coincidence, but due to some unknown or improperly accounted for factors in the heat loss estimation, making these losses about 11 per cent too low for the Grant Street exposure and about 7 per cent too high for the Fourth Avenue exposure.

Improper evaluation of the effect of wind, or improper percentage evaluation of the effectiveness of solar radiation might be assumed to affect these dis-

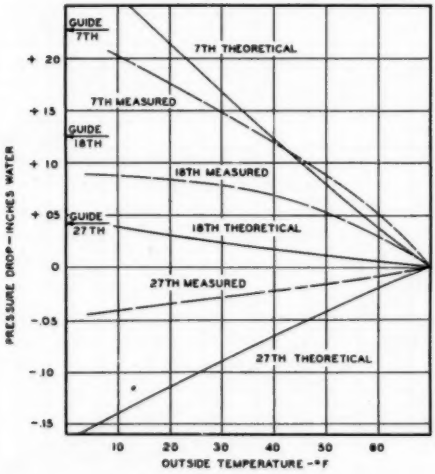

FIG. 7. RELATION BETWEEN OUTSIDE TEMPERATURE AND PRESSURE DROP THROUGH THE WINDOWS ON THE 7TH, 18TH, AND 27TH FLOORS AS MEASURED, AS BASED UPON 100 % CHIMNEY EFFECT AND NO WIND VELOCITY, AND AS BASED ON THE GUIDE FORMULA FOR A 15 MILE WIND AND A 70 F TEMPERATURE DIFFERENCE

crepancies. However, since the estimated heat losses were based upon the observed wind velocity 3 ft outside of the window of each room tested, it is not probable that wind velocity resulted in the discrepancies. Based upon analysis of the data, it was estimated that the sun radiation entering the windows was 50 per cent effective in maintaining room temperatures during the period of the day when the rooms were under test, or from 8 a.m. to 10 p.m. This assumption was largely based upon the southwest exposures facing the direction of maximum sun effect. A closer analysis of the application of this assumption to the northeast and northwest exposures indicates that immediately following sunrise in the morning, and for an hour or more thereafter, radiation entered the windows on the northeast exposure. However, since most of this took place before the beginning of the test at 8 a.m. and resulted in only a negligible amount of heat gain after 8 a.m., little of it was credited in raising the temperature of these rooms. On the other hand, on the northwest exposure, the sun radiation began to enter the rooms at from 2 to 3 p.m.,

depending upon the time of the year, reached a maximum and receded to zero before the close of the test. It might be logical to assume that there is a lag in the effectiveness of the sun radiation so that all of the morning radiation becomes effective in maintaining the desired temperature in the rooms on the northeast exposure during the day, and that much less than 50 per cent is effective in rooms on the northwestern exposure before the close of the test. If it were assumed that all solar radiation entering the room on the north-

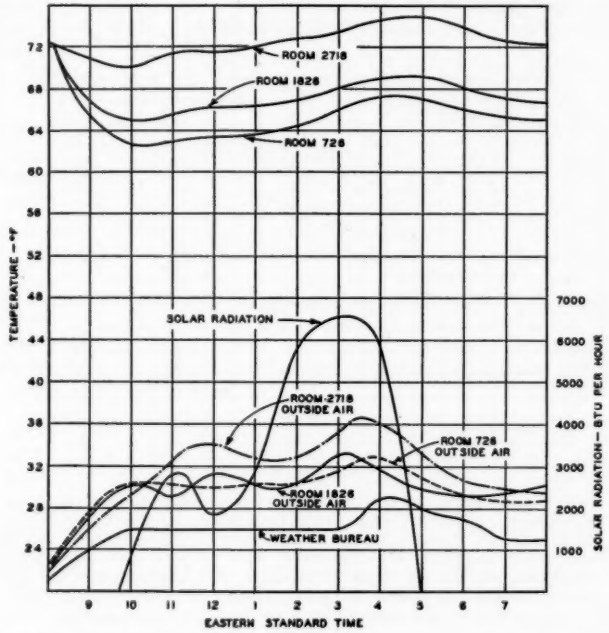

FIG. 8. LOG OF TEMPERATURE CHANGE IN THE ROOMS, 3 FT OUTSIDE OF THE WINDOWS, AND AS REPORTED BY THE WEATHER BUREAU AFTER ALL THE HEAT SUPPLY TO THE ROOMS WAS TURNED OFF AT 8 A.M. SOLAR RADIATION PENETRATION THROUGH THE WINDOWS IS ALSO GIVEN

eastern exposure was effective and none of the radiation entering the rooms on the northwestern exposure was effective, the percentage values given in Column I would be changed to those in Column J, giving better agreement.

While the magnitude of these changes based on these assumptions is not of any great importance for the rooms studied in the Grant Building, they may assume some importance in finally evaluating general assumptions for the effectiveness of solar radiation in rooms of different exposures for design purposes. Thus for a building having exposures facing east, south and west, it may be ultimately proven desirable to assume different percentages for three exposures, with a considerable effect on the estimated heat requirement on

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sunny days. However, since the sun cannot be depended upon to help maintain desired temperatures under most severe heating conditions, it should be observed that sun radiation effects should not enter into the design of the maximum capacity of the heating equipment.

The effect of elevation in the building on heat loss is further emphasized by the results of a test in which all the heat was turned off in the rooms 726, 1826 and 2718, at 8 a.m. The results of this test are plotted in Fig. 8, which gives the room temperatures, the temperatures 3 ft outside the windows, and the Weather Bureau temperatures throughout the day. These three rooms have

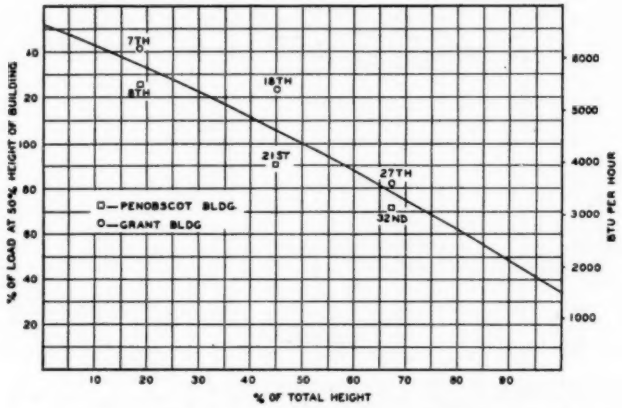

FIG. 9. RELATION BETWEEN THE ELEVATION IN THE BUILDING AS EXPRESSED IN PER CENT OF THE TOTAL HEIGHT, AND THE HEAT REQUIRED TO MAINTAIN THE TEMPERATURE EXPRESSED IN BTU PER HOUR FOR THE AVERAGE ROOM AND IN PER CENT OF THAT REQUIRED FOR ROOMS MIDWAY BETWEEN THE TOP AND BOTTOM OF THE BUILDING, AS OBSERVED IN THE GRANT BUILDING AND AS REPORTED† FOR THE PENOBSCOT BUILDING, DETROIT

the same exposure facing Third Avenue and, hence, are subject to the same solar radiation penetration through the window, which is also plotted. It will be observed that the rate of fall in room temperature was maximum for the 7th floor room and decreased with elevation in the building, and that an approximately constant difference in temperatures was maintained throughout the day. It is also of interest to observe the effect of solar radiation in raising the temperature of all three rooms during the time when the sun was effective and for some time thereafter. It is of interest to note the rise of the air temperatures 3 ft outside the windows with increased solar intensity.

Observations were made on the fall in temperature in rooms on the 4th and 30th floors following the turning off of steam at 9:30 p.m., and when the inside temperature had been maintained at approximately 74 F throughout the day. Between 9:30 p.m., and 8:00 a.m. when the steam was turned on, the average

† Grant Building 424 ft high, 38 stories; Penobscot Building 565 ft high, 47 stories.

rate of fall in air temperature in the middle of a room on the 30th floor was 0.236 F per hour, while the rate of fall for the room similarly located on the 4th floor was 0.571 deg per hour. The results of tests plotted in Fig. 8, and the rate of fall in temperature for the rooms just referred to, are effective indications of the greater heat loss from the lower floors compared with the upper floors in the Grant Building.

The aeration of buildings due to the stack effect is influenced by a number of factors, including the height of the building and the elevation of a given room in the building, the tightness of the building between floors, or the resistance to the flow of air upwards. These are undoubtedly equally effective with the tightness of the windows on the different floors. Also, the cross-sectional area of the building at different elevations affects the pressure gradient. In buildings of the set-back type, where the cross-sectional area decreases with the height, the neutral zone may be expected to be considerably above the middle of the building. This effect is probably more a function of the area of stairways, elevator shafts and such avenues for air flow and the usage of the same, rather than the cross-sectional area. However, since these passage-ways are used by occupants in approximately inverse proportion to their elevation in the building, they may be expected to result in a considerable increase in resistance to flow with height, and therefore, an upward shift of the neutral zone. This is indicated in the *measured* curves in Fig. 7, where the average location of the neutral zone is considerably nearer the 27th than the 18th floor. Actually, the 19th floor is midway between the street level and the top floor served by elevators.

Because of the varied influences of building design, occupancy and operation, on the stack effect, it is necessary to obtain data on a large number of buildings before definite design assumptions can be reached. The curves in Figs. 4, 5 and 6 indicate a proportional average room heat demand as given by the electrical input plus 50 per cent of the solar radiation, corrected to room 1826 exposure, of 5,111, 4,435, and 2,997 Btu per hour for the 7th, 18th and 27th floors, which when plotted against elevation in the building give the relationship shown in Fig. 9. The results of a similar study in the Penobscot Building in Detroit [4] are also plotted. It will be observed that the points resulting from each study form a separate curve, showing different magnitudes of heat requirement and different characteristics of curvature. The different magnitude of heat requirement is a measure of the different exposures of the room studied in the two buildings. The difference in curvature, however, indicates a different relationship between height and heat requirement in the two buildings. The curves are similar in that considerably less heat is required in the upper floors, and the average curve drawn represents fairly well the variation in the percentage requirement in either building. However, it cannot be accepted with great confidence as applying to tall buildings in general, without additional observation in other buildings.

Some such general curve or formula is, however, desired as indicating the general relationship to be expected in buildings of this type, and it is hoped that relatively simple observations in a number of other buildings may be

[4] Influence of Stack Effect on the Heat Loss in Tall Buildings, by Axel Marin, A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 377.

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helpful in arriving at a general formula for the relation between elevation in sky-scraper type buildings to heat requirement, which may be used as a basis for design.

SUMMARY AND CONCLUSIONS

The study in the Grant Building indicates a considerable decrease in heat requirement with elevation in a building, indicating not only a complete absence of heat loss by infiltration in the upper floors, but also an actual decrease in the transmission loss due to the outflow of warm air through windows and walls.

Comparison of the relationship between estimated heat requirement and heat supplied in order to maintain the desired temperatures shows that, when due consideration is given to the actual pressure drop through the window and the effect of solar radiation, the actual input always ranged within 12 per cent, and usually a much smaller percentage, of the estimated requirement. The results of the study in the Grant Building and other similar studies are so varied as to indicate that results on more buildings must be obtained before a general formula for design purposes can be evaluated.

DISCUSSION

C. W. WALTON, JR. (WRITTEN): The following comments are based on three years' observation of heating conditions in the buildings of Rockefeller Center, but not on any such painstaking tests or analyses as have been carried on regarding the Grant and the Penobscot Buildings.

1. By proper control and zoning, considerable steam can be normally saved in the heating of the upper portion of high buildings; however, there are times when wind pressure greatly exceeds the tendency toward exfiltration due to stack action. Tall buildings, generally, have more severe exposures than lower buildings, so that it appears that adequate radiation should be provided on the upper floors to take care of these extreme conditions. This is somewhat analogous to the utilization of sun effect on the south side of buildings during good weather whereby considerable savings can be accomplished if zoning and control permit.
2. On the lower floors (lower quarter or third of building) adequate radiation should be provided for leakage due to combined wind pressure and stack action.
3. Savings can be made by careful attention to open channels such as stairways, elevator shafts, pipe shafts and columns in which piping is concealed.
4. Savings can also be made by careful attention to openings on the ground floor level, such as doors at loading platforms and the use of revolving doors for regular traffic.
5. On the upper floors and lower floors, doors, separating elevator corridors from main corridors, materially reduce the effect of stack action.

No. 1070

AN ALTERNATE METHOD OF COMPARING THE DUST ARRESTANCE OF AIR CLEANING DEVICES

By ARTHUR NUTTING* *(NON-MEMBER)*, LOUISVILLE, KY.

INTENSIVE use of the A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work over a period of five years has indicated it to be probably as practical in the general testing of all types of filters as any single code can be. For the testing of particular types of filters, certain exceptions to the Code could be made which probably would render it more applicable for the filter in mind, but in general the Code as it exists gives a laboratory comparison of air filters very close to their actual field performance. In brief, the present Code prescribes the following procedures and definitions:

1. Dust standardization.
2. Dust feeding method.
3. Rate of dust feed.
4. Air velocity through filter.
5. Method of rating resistance.
6. Method of sampling for arrestance rating.
7. Method of rating filter for dust-holding capacity.

METHOD OF SAMPLING

The method of sampling the dust for obtaining the arrestance rating requires more skill and technique in manipulation than all the other procedures laid down by the Code and at the same time is subject to more possibility of error than any other element of the Code. This sampling is accomplished by drawing through a ¼-in. diameter sampling tube a sample of air on the clean side of the filter. This sample of air is then passed through a porcelain crucible in which all the measurable dust in the air is extracted. The increase in weight of the porcelain crucible is a measure of the filter arrestance. The difficulties of this method are the need for extreme delicacy in weighing, since the increase in weight of the 10 gram crucible is usually in the order of one milligram. Secondly, the sampling tube samples only ⅛ of 1 per cent of the air passing

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Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1937, by H. C. Murphy.

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through the filter and any error in taking the sample will be greatly multiplied. Third, errors in weighing are liable to occur because of the somewhat hygroscopic nature of the porcelain crucible and there is the further disadvantage that as the crucible becomes loaded with dust the sampling rate would be reduced, thereby requiring constant attention for keeping the sampling rate in balance.

It is the purpose of this paper to explain a simple method of determining the arrestance of an air filter. The procedure on which this method is based is not new and has been used in one form or another in many laboratories. To describe the apparatus, reference is made to Fig. 1.

DUST FEEDING METHOD

Air is first passed through a precleaning filter to free it of any impurities existing in the laboratory. Behind this filter a standard feeder, as described in the A.S.H.V.E. Code, introduces a known weight of dust to the air stream

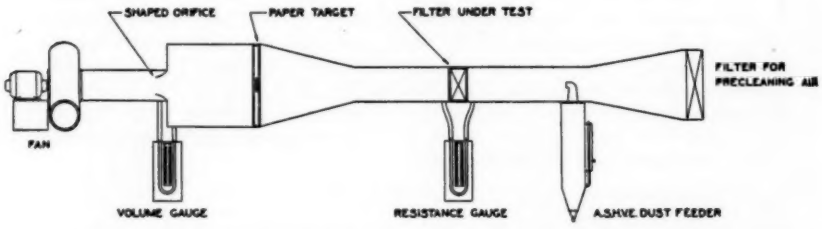

FIG. 1. DIAGRAM OF TEST APPARATUS FOR COLLECTING DUST SAMPLE AND MEASURING RESISTANCE AND AIR VOLUME

at the prescribed rate of feed described by that Code. The dusty air is then drawn through the filter under test and finally through a paper filter target located immediately behind the test filter. The purpose of the paper target is to extract as much as possible of the dust which passes through the filter. The typical methods of measuring resistance and air volume are illustrated in Fig. 1.

When all the dust has been introduced into the test filter, the paper target is removed and placed in the device illustrated in Fig. 2. This device consists of two sheet metal cones mounted with their bases adjacent and arranged so that the paper target can easily be placed between them. At the apex of the lower cone a source of light is mounted and at the apex of the upper cone there is mounted a photo-electric cell. Light passing through the dirty paper target and striking the photo-electric cell will generate a current inversely proportional to the weight of dust on the paper target. If previously a calibration has been made plotting the generated current from the cell against a known weight of dust on the paper target, then it is a simple matter to determine any unknown weight of dust by reference to the calibration curve.

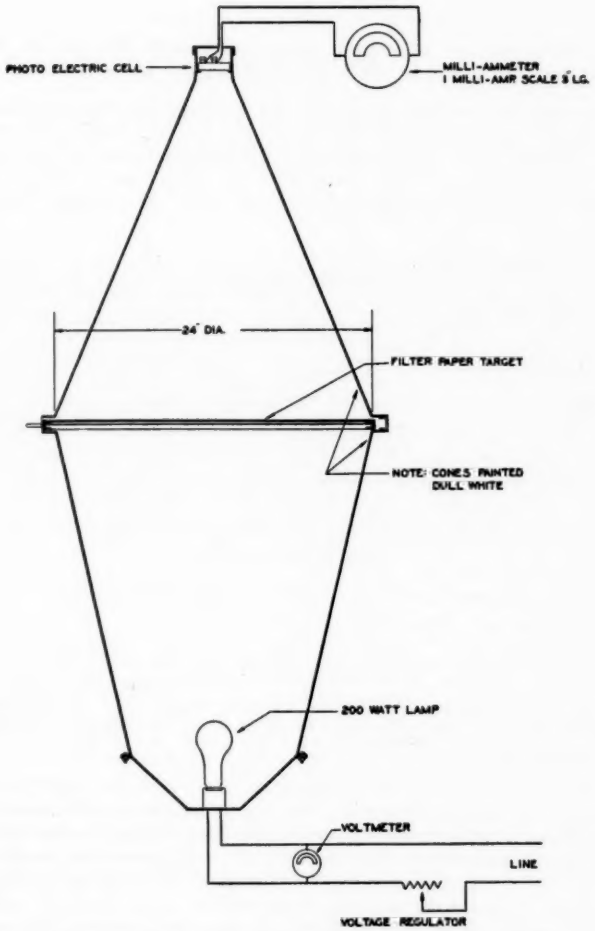

FIG. 2. DEVICE FOR DETERMINING DUST ARRESTANCE

When the type of dust is changed, new curves must be made because of the variable light absorption properties of different dust.

RATE OF DUST FEED

Fig. 3 illustrates two calibration curves made with A.S.H.V.E. Standard Code dust and also using fly ash screened through a 325 mesh sieve.

When using this method of testing filter arrestance, the weight of dust passing through a filter can be determined by a single reading of the milliammeter. Furthermore, the arrestance can be obtained with a relatively small amount of dust fed to the filter instead of the usual 8 or 10 grams required when using a porcelain crucible and sampling tube. This permits feeding the

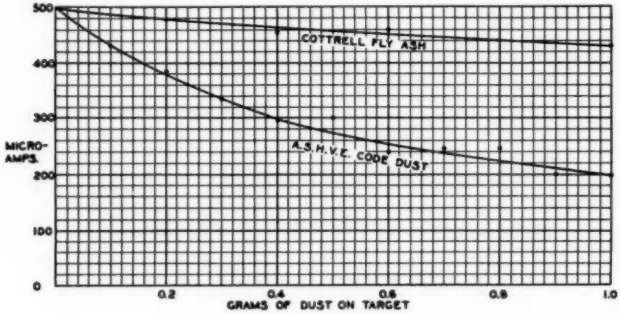

FIG. 3. CALIBRATION CURVES WITH STANDARD DUST AND FLY ASH

dust very slowly, more nearly approaching the dust concentration actually encountered in practical use. The present Code recommends a rate of feed approximately 600 times as heavy as average conditions. In the previously described method of test this rate of feed could be reduced to 100 times average conditions.

One of the chief advantages of this method is that the paper targets can be preserved for future reference. The targets can be visually compared and with the naked eye it is possible to estimate a filter efficiency within 3 per cent. Many of the errors possible with the sampling tube method are largely eliminated by the use of the paper target, chiefly because all of the air and dust from the test filter is passed through the target instead of a fractional percentage. One disadvantage of the target method is that some of the dust may pass through the target, but this is somewhat offset by the fact that when the calibration curve is made, the same quantity of the dust is also lost and the error is balanced. A further disadvantage is that care must be taken that too much dust is not allowed to reach the target, because when dust particles pile one upon the other they act as one particle so far as light absorption is concerned, and error will result.

This method of testing filters has been used for approximately a year and repeated checks between it and the sampling method of the A.S.H.V.E. Standard Code indicate very close agreement when all the factors relative to the sampling tube are free from liable error.

DISCUSSION

PROF. PHILIP DRINKER (WRITTEN): It seems to me that Mr. Nutting's method is far more satisfactory than the original A.S.H.V.E. Code procedure which requires such heavy dust loading of the air stream. As the author points out, this general method has been used for years, notably by J. S. Owens in air pollution work in London. Owens' apparatus has also been used by our Public Health Service in comparing pollution in various American cities (U. S. Public Health Bulletin, No. 224, 1936). Smoky air is drawn for a definite time through a series of ⅛-in. spots on the outer edge of a filter paper disc. The colors of the various spots are matched against standards which represent known dust concentrations.

Mr. Nutting gives no figures and makes no claims as to the accuracy of his method. Judging solely from the proven accuracy of Owens' method, Mr. Nutting's results should be sufficiently accurate for practical filter testing. Inasmuch as Mr. Owens finds it possible to judge the densities of his dust deposits by the simplest of comparison procedures, it is reasonable to assume that Mr. Nutting could do likewise.

R. S. DILL[1] (WRITTEN): The consensus of opinion among those interested now seems to be that the test method for air filters required by the A.S.H.V.E. Code for such devices, however good in other respects, is regrettably inconvenient to use, as Mr. Nutting implies in the first part of his paper.

The new apparatus described in the paper is interesting and its possibilities for the purpose for which it is presumably intended, that is for testing models of filters in process of development, are apparent. Obviously, the apparatus as described is not conveniently applicable to full-sized units.

Experience at the National Bureau of Standards during the past several years indicates the possibility of very practical test methods for air filters based on the use of filter paper for capturing dust and a photometer for evaluating or comparing the amounts of dust so captured.

It appears to us that there are advantages in using filter paper targets on both sides of a filter under test instead of on the clean air side only, and in drawing a sample of air from each side through a target instead of passing the entire discharge through a target.

Such a scheme is applicable to filters of any size. The targets can be smaller than those described in the paper, which were two feet in diameter, and the material used as targets can be dense and therefore of high dust arrestance or efficiency without influencing the power required to force air through the main duct and filter under test.

We feel that a simple and generally acceptable method for testing air filters is highly desirable and that for this reason experimentation such as the author is doing is to be encouraged.

OTTO WECHSBERG (WRITTEN): I am of the opinion that this method is a very great improvement over the present A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices. The weighing of the small porcelain crucibles is

[1] Associate Mechanical Engineer, National Bureau of Standards, Washington, D. C.

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open to so many mistakes that I think it outright unworkable. Even assuming that by proper skill the crucibles can be dried to a standard weight before and after sampling, the slightest scratch made on them in handling will throw the results way off. Notwithstanding the greatest care in handling the crucibles I find again and again that the dust arrestance figured over 100 per cent. I would therefore recommend that Mr. Nutting's new method be embodied as standard procedure in the Code.

The paper states that the photo-electric cell will produce a current inversely proportional to the weight of dust on the paper target. The current, however, is inversely proportional to the shadow produced by the dust particles. For this reason the previously mentioned statement will be true only if the dust particles are of a flaky nature because in such a case the weight of the dust is nearly proportional to the shadow produced. For dust particles of globular shape, the shadow will be proportional to the square of the cube root of the weight.

The calibration of the paper target should be made with slow feeding of the dust and preferably without a test filter installed ahead of the target, so that dust particles of all sizes are caught on the paper target. If the calibration of the paper target is made with an inefficient filter installed ahead of the paper target, the comparison of results obtained with other filters of different dust arrestance would be thrown out of proportion.

Any calibration curve will show that the weight increases faster than the shadow produced on the paper. The critical point is reached where the dust particles start to produce a second layer on the paper target by piling up on top of each other. The calibration curve should be used, therefore, only well below this critical point. Up to the critical point the rate of increase of the shadow should be even though not straight proportional to the weight of dust.

I assume that in the device for determining dust arrestance shown in Fig. 2, the paper target is installed with the dusty side up to prevent some of the dust dropping off from vibration.

R. F. NORRIS[3] (WRITTEN): In our work on the comparison of air cleaners we have come to the same conclusions that Mr. Nutting has reached; that is that the method of sampling the air to determine the efficiency of the air filters as outlined by the A.S.H.V.E. is extremely liable to error. The technique must be carefully followed and, even then, the amounts of dust collected are so small that a small discrepancy in weighing is multiplied so that the results may be seriously in error.

We have used a method similar to Mr. Nutting's in some respects. Our method is to pass all of the air which leaves the filter through a filter paper 48 in. square. This filter paper is then dried and carefully weighed and the amount of dust is determined gravimetrically. This seems to us to be a little less liable to trouble than Mr. Nutting's method where he determines the amount of dust on the filter paper by the transmission of light through the paper. It seems to us that to do this accurately each piece of paper must be calibrated for its light transmission and a correction made in the results for the relative opaqueness of the different pieces of filter paper. It seems to us that the light transmitting qualities of the filter paper can be seriously altered by changes in the moisture content of the paper, since it has been our experience that filter paper which is damp transmits more light than the same paper dry. We do not wish to depreciate Mr. Nutting's work in any way. In fact, we feel that he should be complimented for developing a comparative measuring device which is as simple as this paper shows. We feel, however, that we would rather balance grains fed to the filter against grains passed by the filter rather than grains fed to the filter against milliamps, since there must be some possibility of error in converting grains through the filter to milliamps and back again.

[3] C. F. Burgess Laboratories, Inc., Madison, Wis.

S. R. LEWIS (WRITTEN): My staff has had a wide experience with the Code method of testing air filters. Our feeling is that Mr. Nutting has made a very decided contribution to the process and that the Code should be revised to accommodate this method.

We also feel that the Code and Mr. Nutting's Fig. 1 should take cognizance of the well-proved tendency of dust to settle out of the air stream and be deposited on the bottom of the duct between the filter under test and the target or sampling tube, and that to this end the sampling tunnel should be vertical, so that the entrained dust has a better chance to reach the target.

MARGARET INGELS (WRITTEN): The Alternate Method of Comparing the Dust Arrestance of Air Cleaning Devices presented by Mr. Arthur Nutting is one which to me seems an unusually intelligent and honest effort to answer a very difficult question. How much dust goes through an air cleaning device? Collecting the dust, which passes through a commercial filter, on a chemical filter paper with an accurate means of measuring the dust collected is as sound a method of commercial air filter testing as has yet been devised. The idea of all the air being drawn through the paper eliminates an unknown and probably large error which occurs when only a sample of air from the clean side of the commercial filter is tested. Some colloidal matter does go through the paper, but I believe chemical filter paper is the best filter medium to date and therefore the best one to use as if it had 100 per cent arrestance efficiency.

The same drawbacks for using the method of measuring the dust sample with the photo-electric cell described in the paper occurred in measuring the dust sample with the Anderson-Armspach Dust Determinator where an increased resistance to air flow through the paper was used. The kind and size of dust used for calibration may not be exactly duplicated in tests, and uniformity of dust in the test air may not be constant, so that the dust will be collected at varying rates (being imbedded in the paper at one time and stacked up on the surface at another) effecting final conclusions drawn from a test. A large number of tests as accurately controlled as possible should even out the variations and give honest figures.

I would like to extend my personal commendation to Mr. Nutting for his contribution on the subject.

W. L. FLEISHER: In connection with the paper presented by Mr. Nutting, I feel that the subject, as presented, is of interest, but that this is the time for me to express myself in connection with all dust tests and all dust elimination guarantees.

All of the methods that have been suggested and that have been in use have been quantitative tests rather than qualitative. In the new proposed code or standards for air conditioning which are being developed, dust elimination requirements are based on the elimination of 95 per cent of all particles over 10 microns in diameter. This might have absolutely nothing to do with the quantity, by weight, of dust that either passes the filters or is arrested by the filters. In guaranteeing and specifying dust removal, when we state that a particular filter or any filter will remove 97 per cent or 98 per cent of the dust, do we mean by weight or do we mean by number of particles?

In the elimination of dust to prevent allergies or mould spores, or in the elimination of dust for the prevention of the spread of bacteria or virus, the particle is of much greater importance than the weight. In many cases the smaller particles (under one micron), except in certain cases of silicosis, are of no importance. However, from the angle of soilage or spoilage, the smaller particles could very definitely affect the desired results.

In my own laboratory, with wet filters as used in my capillary conditioners, I had tests made of definite quantities of standard dust introduced into the entering

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side of the capillary filters; and the dust that went through was caught on a very fine filter paper, attached to vacuum cleaner inlets. A similar arrangement was used on the entering side. Although it was not possible to count and size the dust particles on the entering side, due to the vast quantity and the great superimposing of dust particles, it was possible to count and size those on the leaving side. In this way I was able to determine the number of larger particles passing through the eliminator and also their proportion to the smaller size particles, all on a given field. With a microscope of 200 diameters, it was simple to explore a field of about ½ mm in diameter, and fields at various points on the filter paper could be checked in order to get an average of its entire area.

It is my feeling that the Society, in their dust codes, should definitely include some method of qualitative analysis as well as quantitative analysis.

G. W. PENNEY:[3] I believe that those familiar with the testing of air cleaning devices will agree with Mr. Nutting on the difficulties encountered in using the present A.S.H.V.E. Code and will also agree on the difficulty of specifying any single test which adequately measures the performance of all types of air cleaning devices in the various uses for which they are intended. If it were feasible to determine the number of particles of each size in both the ingoing and outgoing air of an air cleaning device and from this determine the efficiency for each size of particle, this would give much more information than our usual test, but the general use of such a test hardly seems feasible with existing equipment.

I think that Mr. Nutting has made an excellent proposal in suggesting the recognition of a test based on the ability of air-borne dust to render filter paper black or opaque.

In industrial air cleaning, to recover valuable material being carried off as a dust, a weight efficiency is a very satisfactory definition of dust removal. However, in air conditioning a weight efficiency is frequently almost meaningless, for by taking out a few large particles, a high weight efficiency may be obtained even though only a few of the particles have been removed. For example, a test using a dust sifted through a 200-mesh screen permits particles up to 0.0025 in. in diameter, but such particles have a volume over 200,000,000 times that of a typical tobacco smoke particle 0.000,004 in. in diameter. If, then, we imagine a dust consisting of one such large particle to 200,000 tobacco smoke particles, the one large particle would weigh 1000 times as much as the 200,000 small particles so that a filter removing only the large particles would show a weight efficiency of 99.9 per cent even though it has removed only one particle out of 200,001. This is, of course, an extreme case since any dust will have particles of intermediate size, but it does illustrate an error which is so great that actual tests by one method may show 95 per cent, while by another method, less than 25 per cent efficiency.

The proposal to use the blackness of the filter paper to measure the weight of dust in the air is questionable because the relation between weight of dust on the paper and photo-electric cell reading varies with the filter paper and the type of dust. Fig. 3 of the paper is an illustration of this variation with type of dust.

For most air conditioning purposes we are interested primarily in the tendency of air to blacken draperies and surfaces. The tendency of the air to render the filter paper black or opaque seems to be a much more satisfactory measure of the performance of an air cleaning device for air conditioning than the weight of dust on the filter paper. Since the efficiency based on the decrease in light transmission of the filter paper is both more reliable and more representative of operating con-

[3] Mgr., Electro-Physics Div., Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

ditions, this is the method which should be specified as at least an alternate method for air cleaning devices for air conditioning purposes.

ARTHUR NUTTING: I appreciate the spirit in which the criticisms and comments on this paper were made. Although I feel that the apparatus described is an improvement in many ways over the Code procedure, at the same time I realize the possibilities of further improvement. This paper was submitted to the Society for the purpose of creating an interest in the development of a better test procedure. No claim could possibly be made that the method presented is ideal. It has obvious disadvantages that have been pointed out, but this method, or an improvement upon it, would certainly present possibilities worth considering.

R. S. Dill's comments do not take into account the fact that the primary purpose of this device is to eliminate the need of sampling the air stream. Therefore, it is impossible with this arrangement to take a target on the dirty air side of the filter. It is true that full size units cannot be tested, and while this would be a disadvantage for laboratories like the Bureau of Standards, it is an advantage to manufacturers who often prefer making their experiments on a small scale.

W. L. Fleisher suggests that filter arrestance be expressed qualitatively as well as quantitatively, and proposes that sizes and numbers of dust particles be included in the qualitative criterion. This suggestion has been made many times before and the adoption of it entails so many complications that I doubt if it will ever come into common practice. In the first place, the particle count results obtained in a laboratory with an artificial dust would in no wise indicate the quality of field service to be expected. Laboratory results on a weight basis, however, are strictly comparable to the weight of dust removed by a filter installation. The reason for this is that measuring efficiency by particle count places as much emphasis on an ultra-fine particle as on a coarse particle. The smallest atmospheric impurities are invariably finer than the smallest particles in an artificial dust and the proportion of fine to coarse particles in atmosphere is many times greater than the same proportion in artificial dust. Millions of these ultra-fine particles passing through the filter would have no appreciable effect on weight efficiency, but in particle count efficiency these ultra-fines would account for practically the entire number of particles passing through the filter.

There is no air filter on the market today, including Mr. Fleisher's capillary air washer, since he mentions it in the discussion, which in terms of number of particles caught will give an efficiency ratio of the magnitude ordinarily expected of efficient devices. Except for special applications, such as bacteria removal, etc., the field of general ventilation is not prepared to pay for, let us say, 70 per cent, 80 per cent, or 90 per cent particle count efficiency down to a half micron size. In general ventilation there is no need or demand for cleanliness of this degree. Ninety per cent particle count efficiency would mean a loss of dust through the filter of about 1/000 of 1 per cent by weight.

Aside from these arguments, the adoption of Mr. Fleisher's suggestion would mean untold labors in the laboratory on methods so tedious and complicated that no two operators could agree. It would further impose a definite specification of the microscopic equipment and its manipulation and would need to define the size limits of the particles to be counted. Further, it would require highly trained laboratory operators, specially skilled in the use of the microscope. There would also have to be designed a sampling method far more delicate than the sampling method now employed.

The U. S. Public Health Service in conjunction with other cooperating organizations has made tremendous efforts to standardize on dust count methods in connection

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with health hazards in dusty occupations. Because of the dissimilarity between the contamination in occupational air and ordinary air, even more effort would have to be expended in the development of equivalent dust count apparatus. Mr. Fleisher's method of counting particles imbedded in filter paper would not do.

No. 1071

CORROSION STUDIES IN STEAM HEATING SYSTEMS

By R. R. SEEBER * (*MEMBER*) AND MARGARET R. HOLLEY ** (*NON-MEMBER*)
HOUGHTON, MICH.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Michigan College of Mining and Technology.

THIS paper describes the progress of research in the corrosion of iron return mains of steam heating systems at the Michigan College of Mining and Technology. The test equipment represents, on a small scale, the ordinary house heating system with boiler, radiator and piping, all of commercial iron and steel.

ARRANGEMENT OF TEST APPARATUS

The test equipment has been described in detail in previous papers [1] before this Society. The arrangements of the parts of the system and points at which changes in testing methods have been introduced are shown in Fig. 1.

Condensate is removed from Loops 3 and 4 and from the return line, and a sample of steam is obtained. These four samples are analyzed daily for oxygen, carbon dioxide, and pH. The standard Winkler method of oxygen analysis is still being used. The pH is determined by means of a quinhydrone set. The carbone dioxide is determined in two ways: (1) by direct titration with barium hydroxide to the phenolphthalein end point (Dr. Guernsey); and (2) by the evolution method as described by Schroeder and Fellows.[2] The latter method involves acidifying a 200-cc sample of the condensate, heating it and driving off all the carbon dioxide, absorbing the carbon dioxide in barium hydroxide and back titrating the excess hydroxide with hydrochloric acid.

The condensate from Loops 3 and 4 is removed by the mercury bottle method as described in the previous paper. The return line sample is allowed to flow

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** Research Chemist, Michigan College of Mining and Technology.

[1] Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman, and G. E. Smedberg, A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, and Vol. 42, 1936.

[2] *A.S.M.E. Transactions*, Vol. 54, R. P. 54-13, p. 213, 1932.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1937, by R. R. Seeber.

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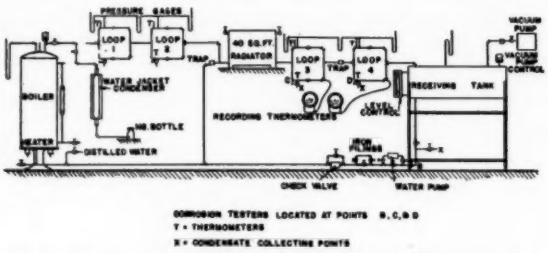

FIG. 1. LINE DIAGRAM OF CORROSION TEST APPARATUS

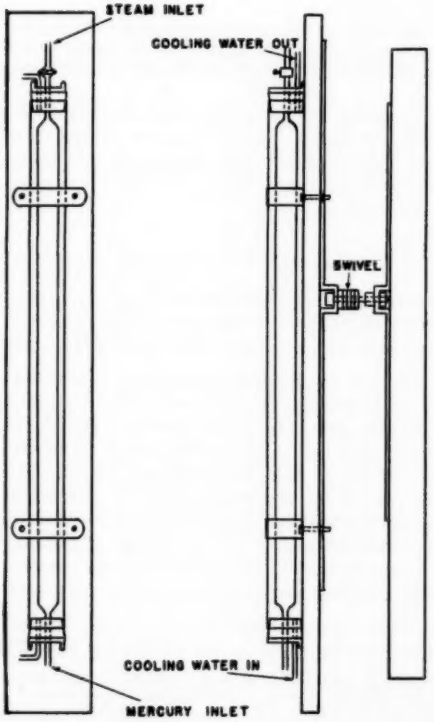

FIG. 2. GLASS STEAM SAMPLING APPARATUS

into a flask which is immediately stoppered. Because there is no gas phase above the sample and because the water runs so rapidly, there is not time for the sample to be affected by the atmosphere.

The method of obtaining the steam sample has been changed. The apparatus consists of a water-cooled condenser mounted on a swivel on the side of the boiler, Fig. 2. The inner section is filled with mercury by raising a leveling bottle connected to the bottom of the condenser. When the condenser is full of mercury and all air is excluded, the clamp is adjusted and the condenser is attached to a petcock on the top of the boiler by means of a rubber tube. The petcock and clamp are opened and the mercury leveling bottle is lowered, thus pulling the steam in from the boiler. The steam is condensed and, when enough has been obtained for an analysis, the petcock and clamp are closed. The mercury bottle is then raised and the condenser is rocked on the swivel to dissolve any gas which may be present. Finally, the apparatus is disconnected from the petcock and the condensate forced out into a flask by raising the mercury bottle and opening the clamp. While this method is superior to others tried, it is still not ideal. To be truly representative of the steam, a sample must condense all the steam and absorb all the gases associated with the water vapor.

A sample of the condensate taken from below the receiver (point *B*, Fig. 1) would seem to most nearly represent the average steam. The water and the gases in solution flow from this point about 6 ft and then enter the boiler, where they form the steam used.

The steam sample taken from the top of the boiler by the glass tube condenser and mercury bottle is analyzed each day for every test.

The first objective has been to relate the corrosion rate (measured in inches penetration per year times 1000) to the oxygen and carbon dioxide concentration in the condensate at the three test points, and if possible to relate these concentrations of oxygen and carbon dioxide in the condensate to the amounts of the same gases present in the steam.

The difficulties to be overcome are due principally to the great number of causes which may affect the corrosion rate. For a system of this sort using distilled water, the following list is believed to include all those causes which produce any appreciable effect on the corrosion rate: oxygen, carbon dioxide, temperature, rate of flow, character of the specimen, pH, diffusion rate and diffusion constants, pressure, and the gaseous phase. By keeping the pressure very nearly constant and the surrounding atmosphere at a constant temperature, it has been possible to hold within narrow limits the variations of the temperature and rate of flow at the various points of the test.

The pH value under the conditions of tests during the past year has varied but little, averaging about 5.5. Since the corrosive action on the test specimen takes place under a film of corrosion product (presumably FeOH) the actual pH at this point is more nearly related to the pH of this corrosion product than to the pH of the surrounding solution until the latter becomes strongly acid (below pH 4) or strongly alkaline (above pH 8).

Speller says: "With waters in the neutral range, the surface film tends to remain at a pH of 9.5 regardless of the pH of the water as there is not enough acid diffusing in from the main solution to neutralize the surface

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TABLE 1—TABULATION OF CORROSION TEST DATA

TEST NO. 61

RETURN LINE				LOOP NO. 4				LOOP NO. 3			
Time Hours	O_2 cc/l	CO_2 ppm	Temp. F	Time Hours	O_2 cc/l	CO_2 ppm	Temp. F	Time Hours	O_2 cc/l	CO_2 ppm	Temp. F
0.3	4.97	11.4	90	1.6	0.89	8.6	153	2.6	0.50	1.4	177
25.5	3.61	12.2	93	28.8	0.89	11.2	153	30.3	0.40	1.0	177
52.8	3.58	12.9	91	54.3	0.99	8.6	153	55.3	0.27	1.4	177
76.1[a]	3.21	12.2	88	77.3[a]	0.79	10.3	153	78.8[a]	0.33	1.4	177
97.3	3.38	12.2	84	97.9	0.93	10.0	153	98.3	0.36	1.4	177
125.3	2.95	12.2	91	126.3	0.70	10.0	153	127.3	0.23	1.4	177
143.8	2.38	11.7	88	145.3	0.33	10.0	153	146.3	0.36	1.4	177
168.1[b]	2.22	12.2	88	172.6[b]	0.53	11.4	153	173.6[b]	0.27	2.1	177
191.8	1.82	11.4	88	193.3	0.56	10.0	153	194.1	0.23	1.4	177
215.7	1.39	11.4	90	217.3	0.46	10.0	153	218.1	0.33	1.4	177
244.3[c]	0.93	10.7	95	246.1[c]	0.27	10.0	153	246.8[c]	0.43	1.0	177

[a] 9 liters H_2O added—6.06 cubic centimeters per liter O_2.
[b] 9 liters H_2O added—1.95 cubic centimeters per liter O_2.
[c] 9 liters H_2O added—1.46 cubic centimeters per liter O_2.

TEST NO. 62

2.0	1.13	10.0	90	3.3	0.99	7.9	152	4.8	0.30	1.4	176
21.0	1.36	9.3	91	23.3	0.76	7.9	152	23.3	0.36	1.4	176
46.5	1.26	7.2	84	46.8	0.50	7.9	152	.....	.....	.....	.....
69.8[d]	1.36	8.6	86	70.5[d]	0.50	7.9	152	.....	.....	.....	.....
97.5	0.93	8.6	91	99.0	0.43	7.9	152	100.3	0.36	1.4	176
121.5	0.73	8.6	91	124.5	0.33	7.9	152	124.8	0.46	1.4	176
145.9	0.50	9.3	90	145.8	0.30	7.1	152	148.1	0.36	1.4	176
170.0	0.60	8.6	91	171.5	0.53	8.3	152	172.5	0.36	1.4	176
189.0[e]	0.33	8.9	93	193.5[e]	0.43	8.6	152	194.8[e]	0.56	1.4	176
214.5	0.36	8.6	84	214.3	0.23	7.1	152	.....	.....	.....	.....
243.0[f]	0.33	8.6	93	243.3[f]	0.40	7.1	152	243.6[f]	0.60	1.4	176
261.2	0.20	8.6	91	262.3	0.27	7.1	152	263.3	0.23	1.4	176
286.8	0.17	7.9	90	287.5	0.20	7.1	152	290.0	0.36	2.1	176

[d] 9 liters H_2O added—3.63 cubic centimeters per liter O_2.
[e] 9 liters H_2O added—1.00 cubic centimeters per liter O_2.
[f] 9 liters H_2O added—0.86 cubic centimeters per liter O_2.

TEST NO. 64

4.0	3.75	2.9	77	5.5	5.05	6.5	91	.....	.....	.....	.....
23.8	5.28	7.9	73	25.0	4.40	8.6	87	5.0	4.86	7.2	105
52.0	5.41	7.2	79	53.3	3.91	8.6	90	30.0	3.89	7.2	107
76.4	5.51	7.2	77	77.5	2.58	7.2	100	54.5	2.51	7.2	120
95.4	4.96	7.2	77	96.6	2.25	6.5	106	73.5	1.79	5.8	133
124.7	4.11	7.2	82	124.7	1.63	6.5	120	101.6	1.04	5.0	147
143.5	3.94	8.6	84	145.0	1.79	6.5	127	126.2	1.34	4.3	156
172.3	3.10	8.3	88	173.8	1.27	7.2	127	150.5	0.72	5.0	157
191.3	2.54	8.6	97	192.3	0.59	7.2	145	169.3	0.49	2.2	170
220.0	2.64	6.5	93	221.3	1.04	7.2	127	198.0	0.36	3.6	152
239.5	2.77	5.8	88	241.5	0.95	7.2	127	218.3	0.68	3.6	156
262.8	2.41	7.2	86	264.8	0.59	5.8	135	239.8	0.49	3.6	170

TEST NO. 51

20.7	2.44	8.8	.....	24.0	2.14	9.5	91	25.0	1.97	9.9	105
45.7	2.49	8.8	.....	48.0	1.93	8.8	98	49.2	1.55	8.8	111
69.5	2.44	8.8	.....	71.0	1.85	10.2	103	77.1	1.51	7.3	119
93.1	2.40	8.8	.....	94.8	1.81	10.2	100	96.0	1.22	7.3	120
142.3[g]	2.11	8.8	.....	147.0[g]	1.72	8.8	95	148.0[g]	1.47	10.2	110
165.9	2.02	8.8	.....	170.8	1.55	10.2	100	172.9	1.43	7.3	117
190.0	2.02	8.8	.....	192.5	1.26	9.1	104	194.0	1.05	8.5	125
212.9	1.85	8.8	.....	214.8	1.09	8.5	105	219.0	1.22	7.3	130

[g] 12 liters H_2O added—6.02 cubic centimeters per liter O_2.

alkalinity. If, however, the main solution is strongly alkaline, the film itself becomes more alkaline, the solubility of ferrous hydroxide is decreased, and the hydroxide is precipitated more rapidly with the formation of a more protective rust coating. This explains the action of alkalies in reducing corrosion. Conversely, if the main solution is sufficiently acid, the acid diffuses in toward the metal rapidly enough to dissolve the protective ferrous hydroxide rust coating and to destroy the alkalinity of the film. Rapid corrosion can then occur, both because the protective coating is not present and because hydrogen gas can be generated from the more acid solution against the metal." [3] Whitman, Russell, and Alterie agree with this.[4]

The introduction of the electrical resistance method of measuring corrosion rates as described in the previous paper [5] presented before the Society, and the

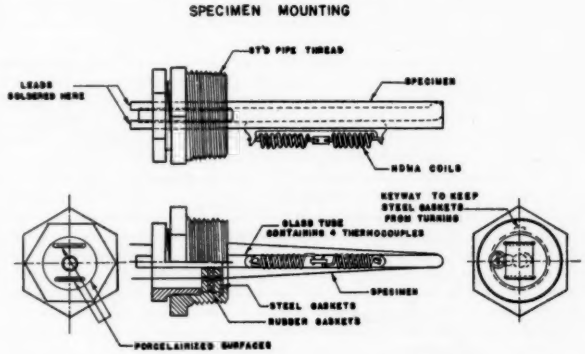

FIG. 3. DIAGRAM SHOWING MOUNTING OF TEST SPECIMEN

American Society for Testing Materials, January, 1937, has made possible a much greater number of corrosion rate observations without stopping the tests to remove specimens. While standard *National District Heating Association* test coils have been exposed in each test in the same location as the resistance test strips, for this discussion only electrical resistance rates are used. The mounting of specimen is shown in Fig. 3.

The testing system is a closed system: the condensate returns to the boiler, and the only change in the liquid contents is caused by the removal of samples for chemical analysis and the occasional addition of water to the boiler. The system is practically tight. Before starting a test a high vacuum is produced, which changes only a fraction of an inch in 24 hours. All tests from which data (given in Table 1) for this paper were taken, were at atmospheric pressure so that the gaseous phase was changed only by the progress of the test.

The test strips are made from cold-rolled Bessemer strip steel furnished by

[3] Speller, Corrosion, Causes and Prevention, 2nd Ed., p. 162.
[4] Whitman, Russell and Alterie, *Ind. & Eng. Chem.,* Vol. 16, p. 665, 1924.
[5] Loc. Cit. See Note 1.

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the U. S. Steel Co. through their Cuyahoga Works at Cleveland; it has been analyzed by them as:

Carbon	0.10
Manganese	0.30
Sulfur	0.037
Phosphorus	0.077

or almost exactly the analysis of the *National District Heating Association* test coils. Although these test pieces are prepared by exact routine with an annealing by electricity in a vacuum for each, it is not to be expected that the

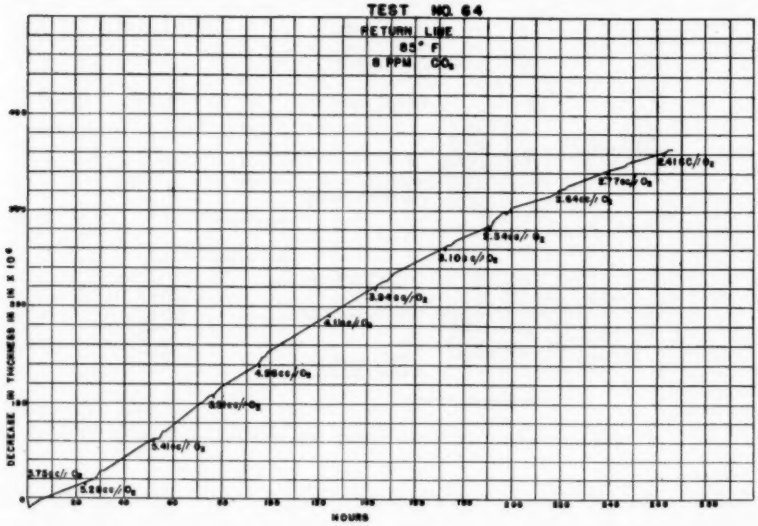

FIG. 4. CURVE SHOWING DECREASE IN TEST STRIP THICKNESS AND OXYGEN ANALYSIS OF SAMPLES IN RETURN LINE

resistance to corrosion of all pieces is precisely the same. Many tests under similar conditions should indicate the mean. The unit resistance of these specimens varies but little.

The decrease in thickness of the test strip with time is given in Fig. 4 and also the O_2 analysis of samples taken at the various times indicated by the position of the analysis on the chart. This plate gives results from one test on condensate from the receiver and the analysis represents most nearly the chemical constituents of the steam at the times given.

Fig. 5 charts similar information obtained from Loop No. 4, when a lower temperature than usual was desired and it became difficult to maintain this temperature at a constant figure. The temperature variations were therefore plotted to scale on the chart. Similar charts were prepared for each

test, and from the corrosion rates obtained from these and the corresponding chemical analyses, Figs. 6 and 7 were plotted.

Most of this work was done with CO_2 concentrations of about 10 parts per million. Similar work must be done with other concentrations of CO_2 to enable deductions to be made as to the probable effect of changes in this constituent.

Assume a test starting with 1.0 cc per liter of O_2 in the condensate in the receiver (test point 3). As the test proceeds, this oxygen is reduced by combination with the iron of the system and approaches a value of zero cc per liter. If samples for analysis are taken at intervals and a curve is plotted of reduction in thickness of test specimen against time (Fig. 4), a line con-

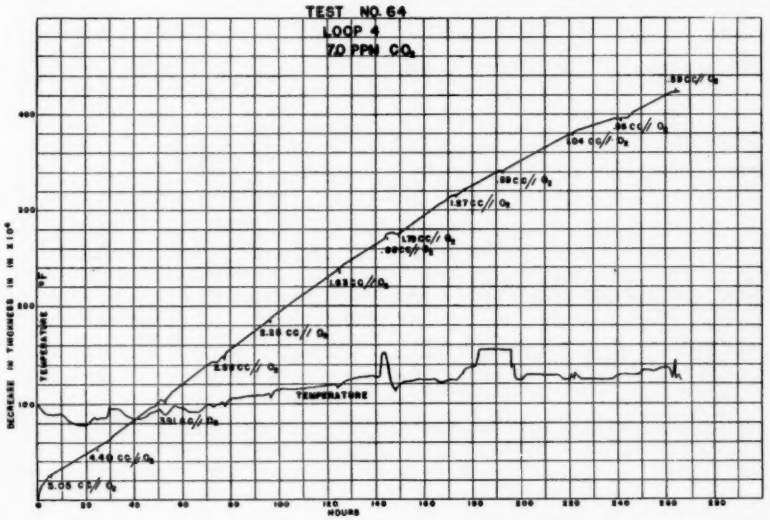

FIG. 5. CURVE SHOWING DECREASE IN TEST STRIP THICKNESS AND OXYGEN ANALYSIS OF SAMPLES IN LOOP NO. 4

necting two consecutive points of sampling (shown at intersection of sampling time with curve) will show the rate of decrease of thickness and consequently the inches penetration per year, while the average of the two analyses will indicate the oxygen concentration related to this rate.

From plotted results obtained in this manner for various O_2 concentrations a curve showing change of corrosion rate with change of O_2 concentration may be approximated. In tests thus far this year CO_2 was also present. Where the temperatures are not too great, as in Loop 4 and in the return main, many tests have shown consistently that, while the oxygen decreases with lapse of time, the CO_2 concentration remains substantially constant. As the CO_2 remains constant the only variable is the oxygen, and curves can be drawn for the corrosion rate of O_2 in the presence of a given amount of

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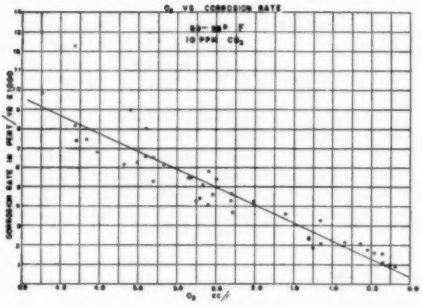

FIG. 6. (*Left*) CORROSION RATE PLOTTED WITH REFERENCE TO THE OXYGEN CONCENTRATION OVER A TEMPERATURE RANGE FROM 80 TO 95 F

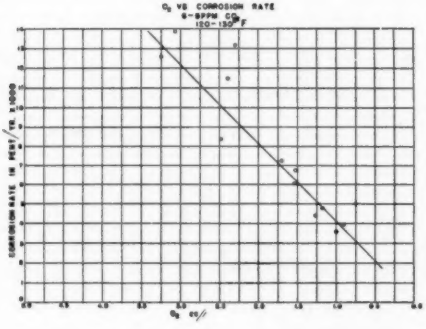

FIG 7. (*Right*) CORROSION RATE PLOTTED WITH REFERENCE TO THE OXYGEN CONCENTRATION OVER A TEMPERATURE RANGE FROM 120 TO 130 F

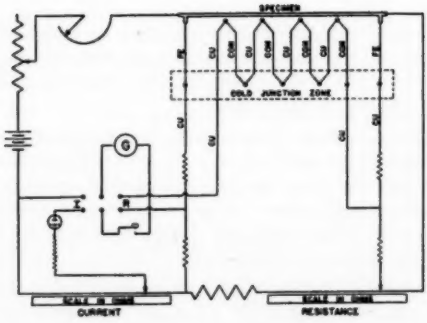

FIG 8. (*Left*) COMPLETE ELECTRICAL CIRCUIT DIAGRAM FOR MOUNTING OF A RESISTANCE SPECIMEN

CO_2 and at constant temperature and at a definite rate of flow of condensate past the test specimen.

The test results plotted (Figs. 6 and 7) indicate that the corrosion rate increases directly with the O_2 concentration, CO_2 being constant. With the same concentrations of gases and higher temperatures, the corrosion rates are much higher.

Although the plotting of test results shows wider individual variation from a mean than is desirable, the inferences drawn appear to be justified. These variations in results seem to have been caused by actual variations in some of the many variables which it was attempted to maintain constant throughout the test periods. While the methods of analysis are quite accurate, it is unlikely that the samples taken are at all times truly representative of the liquid in contact with the test specimen.

One interesting step will be to reduce the O_2 concentration to zero and then note the effect of the CO_2 acting alone.

Although definite information as to the effect of temperature change on the rate of corrosion with constant O_2 and CO_2 concentration in the condensate can be expected from a continuation of this research, this information might be more rapidly obtained from a laboratory testing equipment such as used by Finnegan, Corey and Jacobus.[6]

The effect of variations of rate of flow or velocity of the stream of condensate passing the test specimen should be carefully investigated. This research might well be combined with the investigation of the effect of temperature changes.

The electrical resistance method of corrosion testing has been improved during the past year. Variations in observed resistances were found to be due to variations in contact resistance between the leads. A new method of making these contacts reduced variations materially. A new mounting of the resistance specimen as shown in Fig. 8 reduced troubles from possible contact of resistance strip with *National District Heating Association* testers or the plug itself.

ACKNOWLEDGMENTS

Acknowledgment is due to G. H. Damon of the Chemistry Department, Michigan College of Mining and Technology, for much advice and assistance, and to G. H. Lyle of the Electrical Engineering Department, Michigan College of Mining and Technology, for assistance in improving the equipment for electrical resistance measurements.

DISCUSSION

THOMAS FINNEGAN[7] (WRITTEN): Those who are concerned with the utilization of large quantities of steam are eager to learn all that they can about corrosion in steam systems. The utility companies engaged in the selling of steam realize that they must gain all the information possible in order to maintain the integrity of

[6] *Ind. Eng. Chem. 27*, p. 774, 1935.
[7] Chemical Div., Engrg. Research Dept., New York Steam Corp., New York, N. Y.

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their service. As a result there has been considerable research work performed during the last decade, both in the field and in the laboratory.

Professor Seeber has combined these two phases in his work. He is performing a laboratory study in that he is seeking information regarding the fundamentals of corrosion. It is a small-sized field installation in that actual steam utilizing equipment is being employed. If successful results can be obtained with this apparatus they will be valuable, because there is a certain hesitation on the part of the more practical-minded steam users to accept as final the results of pure laboratory studies, although the more scientific fraternity realizes that the fundamentals of aqueous corrosion are common to any installation, providing the problem is properly defined.

There are great difficulties in controlling the variables under study when a real heating system is used, so Professor Seeber must be commended on his courage in starting this work.

Interested persons will naturally ask the question, "What have we found out regarding heating system corrosion when we study the present paper?" It must be reviewed in this light.

I shall not concern myself with more than passing comment regarding the electrical method of measuring corrosion. It is presumed that it has been tested in comparison with the more conventional methods of using weight loss data, and has been found to be comparable, and to give reproducible results. If such be the case, attention may be immediately directed to the results of Figs. 6 and 7 of Professor Seeber's paper.

These figures present two lines; one for an average of 7 ppm of CO_2, the other for 10 ppm of CO_2. A comparison of the curves will show that for the same oxygen concentration, a higher corrosion rate is found for the lower CO_2 than for the higher. This is indeed surprising.

It is surprising because CO_2 in solution will lower the pH value of the water. It is the opinion of many, including this discussor, that oxygen is a more influential factor than CO_2, but I doubt if anyone has hitherto believed that CO_2 is not, by virtue of its acidic character, a factor of some importance. One of the reasons why the district heating industry has largely discarded the use of soda ash in boiler treatment is to minimize the CO_2 content of the steam. These curves, at first glance, would indicate that such procedure is unnecessary.

To be fair one must study the curves further.

The ordinary methods of determining CO_2 are not as precise as the method used to determine oxygen. It may be, then, that we can consider the CO_2 in both curves as practically the same. If this be done we have temperature as the variable with Fig. 6 representing 88 F as the average of the extremes, and Fig. 7 representing 125 F. The curves then indicate that temperature becomes a highly important factor when the oxygen increases. Both curves approach a corrosion rate less than 0.001 in. per year as the oxygen approaches zero, suggesting that temperature alone is not highly effective, but as we increase the oxygen above about 1 cc per liter, the corrosion rate is approximately twice as high at 125 F as it is at 88 F. In other words, it would appear that if it were inconvenient to decrease the oxygen content of a condensate, corrosion protection could be achieved by cooling the condensate.

It is a general statement that chemical reactions take place more vigorously at higher than at lower temperatures. But it is surprising that the temperature coefficient of dissolved oxygen corrosion should be such that warming the condensate from about room temperature to some 40 deg higher will double the corrosion rate, as is suggested by Figs. 6 and 7.

I say suggested, because one cannot say that any principle is established by a

single change of the variable. One would require a large group of curves in which CO_2 varied from zero to practically maximum value, and temperature ranged in fairly short steps throughout the entire region which is encountered in practice.

We do not question that temperature is a factor. We want, however, quantitative data which will be applicable when we are called in to assist in the elimination of corrosion in a heating system. We have, perhaps, received a ray of light with the present publication, but we have not received enough illumination to clear up the darkness which we find in the pipe shafts and cellars of the large buildings which use steam for such a multitude of purposes.

Figs. 6 and 7, which in combination formed the climax of the paper, tell us that oxygen has a quantitative effect and that temperature is an important factor. Unfortunately, they do not tell us anything about the effect of CO_2. This is particularly disappointing because whenever chemists and engineers talk about heating system corrosion among themselves more discussion and debate are given to the question of the effect of CO_2 than to that of any other corrosive factor.

For the present we must set aside the question of CO_2 and content ourselves with the fact that we have some data in Figs. 6 and 7 to indicate the quantitative effect of dissolved oxygen.

I was not able to construct the plots of Figs. 6 and 7 from those of Figs. 4 and 5, so I am forced to conclude that the latter are purely for illustrative purposes, and that Figs. 6 and 7 are prepared from other data which are not in the paper. I was not even able to construct a similar plot from Figs. 4 and 5 because an inspection of the curves shows that they deny the quantitative effect of dissolved oxygen.

Corrosion rate is expressed as decrease of thickness per time. The slopes of the curves in both Fig. 4 and Fig. 5 are therefore corrosion rate. These curves are practically straight lines, or at least there is very little curvature. The slope is therefore practically constant, which is the same as saying that the corrosion rate is practically constant. This is in spite of the fact that there is a continual decrease of oxygen throughout the test period.

It is not possible to determine, or to imagine, why this is so. It may be the result of a change in the character of the specimen as the lower oxygen concentrations are reached. It may be questionable technique in measuring the corrosion. It is obvious, however, that Figs. 4 and 5 do not establish the relation of Figs. 6 and 7 unless there is some mathematical device which can be used, which this discussor has never been fortunate enough to learn.

Another matter which demands clarification is the magnitude of the oxygen results found at Loop 4. This point is after the trap and the water passing is the condensate formed in the radiator. If the trap is working properly no steam vapor should be condensed in the sampling apparatus as the experiments were conducted at atmospheric pressure.

It will be seen by reference to the table that many of the oxygen results at Loop 4 are of the same magnitude as those of the feed water, which is taken as the composition of the steam. Considering that oxygen, like any gas, is soluble to an extent depending upon its partial pressure, and considering that steam with an equivalent of 5 cc per liter of oxygen has an oxygen partial pressure of only 0.00005 lb per square inch absolute, it is obvious that the condensate formed will have only a minute quantity of oxygen. It may be approximately calculated to give the figure 0.00005 cc per liter.

The fact that the condensate shows oxygen values so much higher than this figure indicates one of several possibilities. For instance:

1. Air in-leakage near loop.
2. Carry-over of boiler water into steam system.
3. The sample represents material which was in vapor state inside the loop.

Considering the low temperature at the loop which, from Fig. 1, appears to have been measured by inserting a recording thermometer into the loop, the first of these possibilities seems the most probable.

It might be mentioned parenthetically that if one is interested in the subject of oxygen partial pressure and solubility, it has been described at length by Hall and Mumford.[8] It might also be mentioned that at 100 F the solubility of oxygen from atmospheric air is only 4.7 cc per liter, according to Winkler, who is a recognized authority on this subject. This figure is exceeded by some of the points on Figs. 4 and 5.

The question then arises: Can such an apparatus be used in a study of steam system corrosion? Can it be operated to give conditions of physical equilibrium so that the concentrations of the dissolved gases in the condensate can be related to those in the steam? Apparently not, unless the laws of Henry and Dalton are wrong! They have been accepted as fundamental scientific principles for years.

We frequently find high oxygen concentrations when we analyze condensate, but we know that they are the result of air in-leakage. No other explanation is possible. We do not know just how corrosive uncontaminated condensate might be, and we would like to be informed.

In view of the results as they have been presented so far I can only say that they are too confusing to be acceptable. There is a lack of agreement between the two types of graphical presentation. There are unexplained magnitudes of oxygen concentration. A complete re-definition of the problem is in order so that the limitations of this type of apparatus can be definitely established, and so that secondary conditions, such as air in-leakage, may be consciously controlled if desired.

I have not presented this discussion with the intention of being disparaging. I speak, however, as a representative of the industry that is engaged in the distribution of steam. As such I am forced to be highly critical of any work on the subject of steam system corrosion, whether performed by investigators within that industry or on the outside. We are eager to learn the facts, and we have no fear of learning them through public presentation. We have the duty, however, of demanding that the information received be capable of definite interpretation, so that those who experience the annoyance of steam system corrosion will have the necessary background for the study of their problem without any misunderstandings or risk of becoming prejudiced.

C. A. DUNHAM (WRITTEN): Corrosion studies in their relation to steam heating for buildings present a complex problem in research. The fact that there are at least 18 known variables, any one of which under certain conditions of practice might dominate the situation, adds to the complexity. Obviously therefore to make progress, studies must be confined to specific uses in steam heating to meet common practice; and as this knowledge is brought out in the open and made available for general use, then explorations may be made in the less active field of heating.

I feel keenly the importance of all research work being directed into channels where the knowledge gained may be made applicable to meet the most pressing demands, and that the less important phases of research where endless work may be required, be made secondary.

[8] Some Fundamental Considerations of Corrosion in Steam and Condensate Lines, by R. F. Hall and A. R. Mumford (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 121).

The real problem to my mind and one of immediate importance, involves the heating practices of the past now reflected in thousands upon thousands of buildings, which represents the greatest capital loss per year from heating system corrosion.

Present day heating practice, however, is the corrosion problem of tomorrow, and, since this practice involves better and more effective steam circulation, if when correctly understood and followed in practice, I am prepared to say it will materially minimize future corrosion. It seems logical therefore to give precedence in research to the new and better method in heating so that the corrosion corrective measures may be applied even to antiquated systems through modernization, thus inheriting these newly discovered values in corrosion preventatives.

The findings of our own research work in this field have been partially confirmed by Professor Seeber. The expulsion of fixed gases from the system of heating, with a definite flow of steam at all times from any pressure point above absolute to a lower pressure where non-condensable gases are drawn out, has been surprisingly helpful in reducing corrosion in systems where unusual corrosion existed before; probably for reasons that sub-atmospheric methods of circulating steam have made unpopular the laxity in making up the pipe joints. Steam systems can and are now being installed free from the usual air leaks to a remarkable degree of tightness.

Professor Seeber should be encouraged to continue his studies both through financial and moral support; it is my belief that in his search for the *cause* he will uncover the remedy through a study of the *effect* of certain experiments. I suggest three things for active exploration in this field:

1. Study the effect of corrosion where steam is intermittently turned on and off the heating system where atmospheric air replaces steam alternately as compared with the corrosion effect of steam flowing into the system in graduated amounts but continuously.
2. Determine the corrosion effect in steam heating systems at sub-atmospheric pressures in the higher range of vacuums where the system under observation has been installed by fixed standards for air leak detection.
3. Test for corrosion effect of using the standard boiler compounds in a vacuum return line heating system in varying quantities upon both piping and copper alloys.

It is suggested that all these studies be made as close as possible to actual practice operations.

F. L. LAQUE[9] (WRITTEN): The method chosen by these investigators to study corrosion in condensate return lines in steam heating systems should yield results of practical value, since it provides for exposure of test specimens under conditions very similar to those encountered in operating equipment. While it is possible to set up an apparatus so that it will operate under a certain set of conditions, it often follows that the more closely the corrosion testing apparatus is made to approach actual operating equipment—as in the present case—the more difficult it is to make the apparatus flexible enough in its operation to provide for controlled variations in the several factors that influence the corrosion of the material being studied. The authors recognize this difficulty in their suggestion that laboratory testing equipment, such as used by Finnegan, Corey and Jacobus,[10] is better suited than that described in their present paper for studying the effects of such variables as temperature, rate of flow, etc., on the corrosion of iron by condensate containing constant percentages of O_2 and CO_2.

In the course of an investigation of a similar corrosion problem, W. A. Wesley, G. L. Cox, H. R. Copson and their associates in the Research Laboratory of The International Nickel Company, Inc., at Bayonne, N. J., devised an apparatus by means of which it was possible to control, and vary at will, such factors as tem-

[9] Development and Research Division, International Nickel Co., Inc., New York, N. Y.
[10] *Ind. Eng. Chem.*, Vol. 27, p. 774, 1935.

perature, pressure, motion of test specimens, and CO_2 and O_2 concentration. They have kindly given the writer permission to describe their apparatus and report some of their results on iron as part of this discussion.

The corrosion chamber consisted of a pot about 8 in. in diameter by 12 in. deep fitted with a pressure tight flanged cover, which could be bolted on. The test specimens were mounted on a central shaft, which passed through the cover. This shaft was driven by an electric motor at the desired speed, *e.g.*, 16 rpm. The specimens were insulated from the shaft and supporting arms by means of glass rods and porcelain washers. The desired concentrations of dissolved gases were obtained by bubbling a mixture of CO_2 and air through distilled water in the pot. The mixed gases entered the pot through a tube which passed through the cover; the end of this tube was covered with a perforated plate so as to break up the gases into small bubbles to insure saturation of the water. The mixed gases left the corrosion chamber through a water cooled condenser, the rate of flow of the gases being regulated by means of a needle valve on the condenser outlet. The pressure was controlled by a reducing valve attached to the cylinders in which the mixed gases were supplied under a pressure of about 210 P.S.I. The corrosion chamber was fitted with a pressure gage and thermometer, and a flow meter and gas filter were placed in the line between the cylinder and the testing apparatus. The testing chamber was surrounded by an electrically heated oil bath, the temperature of which was controlled by means of thermostats. It was found desirable to use the alloy Inconel for all parts of the apparatus which came into contact with the testing solutions.

With this apparatus it was possible to control the following variables:

1. Concentrations of CO_2 and O_2 by controlling the composition of the mixed gases in the pressure cylinders.
2. Pressure, by means of the reducing valve in the gas line between the cylinders and the testing chamber.
3. Temperature, by means of the oil bath surrounding the corrosion chamber.
4. Velocity, by means of the mechanism used to drive the spindle on which the specimens were mounted, or by the location of the specimens on their supporting arms with reference to their distance from the center of the drive shaft.

The apparatus described was not used for an intensive study of the corrosion of iron by water containing CO_2 and O_2, but some tests were made which may be of interest for comparison with those reported by Seeber and Holley. Some of these results are tabulated in Tables A and B.

TABLE A—RESULTS OF EXPOSURE OF INGOT IRON TO WATER SATURATED WITH MIXTURES OF CO_2 AND AIR AT 70 C (158 F) AND 35 LB GAGE PRESSURE

PER CENT CO_2 IN GAS BUBBLED THROUGH WATER[a]	CO_2 CONC. IN CC PER LITER[b]	O_2 CONC. IN CC PER LITER[b]	CORROSION RATES IN INCH PENETRATION PER YEAR × 1000	
			1st–2 hours	2nd–20 hours
0	0	11.5	53	51
40	381	6.9	430	102
50	476	5.8	270	85
60	571	4.6	280	88
80	760	2.3	248	92
100	952	0	163	59

[a] By volume — balance air.
[b] At 0 C and 760 mm pressure—Calculated using Henry's Law.

TABLE B—RESULTS OF EXPOSURE OF INGOT IRON TO WATER SATURATED WITH A MIXTURE CONTAINING 80 PER CENT BY VOLUME OF CO_2 AND 20 PER CENT AIR AT 35 LB GAGE PRESSURE

TEMPERATURE	CORROSION RATE IN INCH PENETRATION PER YEAR × 1000	
	1st–2 hours	2nd–20 hours
30 C (86 F)	184	76
70 C (158 F)	250	92
100 C (212 F)	450	115
120 C (248 F)	336	30
135 C (275 F)	362	11

It should be noted that the concentrations of dissolved gases achieved in these tests were considerably higher than those reported by Seeber and Holley from their analyses of condensate in their tests. This probably accounts for the higher rates of corrosion observed in these tests, especially in the case of the original rates of attack as measured by the two-hour tests which, under some conditions, were more than 30 times greater than the highest rates observed by Seeber and Holley. Observations of iron and other metals by steam in chemical process equipment rather than heating systems have indicated that the higher rates reported here are not uncommon. It may be noted also that analyses of non-condensable gases in steam have shown carbon dioxide contents of such gases as high as 80 per cent by volume.

From Table A, it would appear that at 70 C (158 F) considerable corrosion may be caused by air alone, or carbon dioxide alone, and that of the two carbon dioxide is the more dangerous. However, there is some evidence, from other tests, that at higher temperatures (248 F) corrosion by air alone may be more serious than by carbon dioxide alone, or mixtures of carbon dioxide and air, and, therefore, it is more important to keep air (oxygen) out of the system than to eliminate carbon dioxide.

From Table B, the conclusion might be reached that corrosion as measured in the second 20 hours of exposure tends to increase with temperature up to about 100 C (212 F) and then decreases sharply. It was evident that corrosion rates decreased considerably with time, probably due to the formation of films of protective corrosion products.

These conclusions are not supported by sufficient data to be reliable, but are offered merely for consideration in the light of practical experience and for comparison with other available data.

If Professor Seeber and Miss Holley should be interested in the possibilities of using a testing device of the sort described here in further studies of this corrosion problem, the writer and his associates will be glad to supply complete detailed information on the design, construction, and operation of this apparatus.

In any event, the writer hopes that these investigators will continue their excellent work and, if possible, supplement their laboratory tests by tests on specimens exposed in operating heating systems in locations where corrosion is known to be serious. It would be interesting to have a direct comparison of the results of the laboratory tests with those of field tests in order to determine whether the rates of corrosion observed in the laboratory are of the same order of magnitude as those observed in the field.

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PROF. D. D. JACOBUS [11] (WRITTEN): Figs. 4 and 5 represent typical data, the run selected being Test No. 64, where Fig. 4 gives the conditions existing in the return line and Fig. 5 gives the conditions in Loop 4. The total decrease in thickness of a test piece is plotted against the time in which the decrease in thickness took place. The oxygen concentration which prevailed at the time an observation was made is recorded as a point on the plotted curve. For the sake of brevity I will confine my discussion to Fig. 5, as the plotted curves are rather similar in character and magnitude.

The slope of a tangent to the curve shown in Fig. 5 represents the rate at which corrosion is taking place. Selecting typical points we find the following:

TIME Hours	OXYGEN cc/l.	SLOPE OF CURVE (in. $\times 10^6$/hour)	CORROSION RATE (in. $\times 10^3$/year)	AVERAGE TEMPERATURE F
26	4.40	1.80	7.9	100
126	1.63	1.65	7.2	130
220	1.04	1.10	4.8	130

If we take these sample observations and attempt to locate them on Fig. 7, we find that there are no corresponding points which correlate with the test data. To be more general, if the curves shown in Figs. 4 and 5 were straight lines it would indicate that oxygen had no effect on the corrosion rate. As a matter of fact the lines shown in Figs. 4 and 5 have slight downward curvatures, indicating that the corrosion rate is decreasing as the oxygen in the apparatus is being used up, but the decreases shown by the data are not nearly as rapid as Figs. 6 and 7 indicate.

I have pointed out that the data which Professor Seeber has plotted in Figs. 4 and 5 are in disagreement with the test results which are recorded in Figs. 6 and 7, and trust that this apparent discrepancy can be overcome by an explanation of the manner in which typical points on Figs. 6 and 7 have been secured.

F. N. SPELLER (WRITTEN): Professor Seeber's experimental work as summarized in his paper has resulted in fundamental data of basic value, confirming with satisfactory precision other data as to the effect of oxygen content, and CO_2, these being the main factors involved in corrosion of steam lines. The method of measuring corrosion under these conditions is novel and makes it easy to obtain the many determinations required to draw time-corrosion charts. This useful laboratory-controlled research should be continued.

One or two low-cost alloy steels and a few protective coatings are now available which may be economically useful for protection of the inside of steam return lines. It seems desirable to carry out a test for relative corrosion and the resistance of such coatings. The factors above mentioned should be present in about the same proportion as is found under average service conditions. In fact, a variation in the oxygen contents, carbonic acid, temperature, and flow-velocity during the test, may be important when the object is to determine the relative corrosion, as the resistance of the metal is dependent upon the stability of its surface film. Moreover, the finish of the pipe metal is also a factor, so that there seems to be good reason for making such tests in the form of pipe in an installation that includes all the variables that occur in service.

[11] Dept. of Chemistry, Stevens Institute of Technology, Hoboken, N. J.

R. M. PALMER[12] (WRITTEN): This paper discloses a painstaking approach to this complex problem. The analytical methods which have been chosen appear to be the best available at this time. As is pointed out in the paper, the study now being made must necessarily be circumscribed in several respects.

It is interesting to note that the aggressive action of carbon dioxide is to receive careful attention. In the past the action of carbon dioxide has been overlooked or ignored. Certain researchers, however, have plainly indicated the important rôle played by carbon dioxide in corrosion problems.

It is my considered belief that parallel tests carried out in a building supplied with district steam would be of the utmost value. Correlation of results obtained in these two sets of tests would make either one of them of greater value.

The most serious corrosion problem encountered in steam heating systems arises when condensate is wasted from the system and not returned to the boiler. It is apparent, therefore, that a thorough test of such a system would be of the utmost value.

Operators of buildings using district steam have encountered widely varying degrees of corrosion in condensate lines.

My attention has been called to the operation of two large office buildings across the street from each other, both using the same district steam supply. In the case of one building most serious corrosion troubles, with actual failure of condensate lines, were experienced at the end of one season's operation. In the case of the other building, no serious corrosion difficulties have been encountered after an operating period of some 10 years.

The operating engineer of the building where no appreciable corrosion difficulty was encountered accounts for the favorable conditions as due to the fact that he invariably exhausts his system by means of a vacuum pump before he permits steam to enter the system from the street. This is probably not the sole factor in accounting for the difference in rates of corrosion encountered. Only a careful survey of every possible variable in the two systems would afford a complete picture of reasons for serious difficulties encountered in the one case and its absence in the other.

It might be stated in passing that substitution of non-ferrous metal piping for ferrous metal has not afforded a satisfactory solution for this problem. It is known that carbon dioxide attacks copper and the brasses. If carbon dioxide were absent from condensate, it is likely that the use of a good grade of brass piping would prove satisfactory from the corrosion standpoint. The oxygen present in the condensate would not cause serious corrosion of brass alone.

During the past decade there have been marked increases both in operating pressures and operating temperatures practiced by district steam heating companies. A study of the amount of dissociation encountered under varying practical operating conditions of pressure and temperature, together with a careful study of composition of steam resulting therefrom, should prove of value in connection with this study of corrosion.

It is my considered belief that this problem of corrosion of steam condensate lines will only be solved when carefully planned and controlled research work is carried on in one or more buildings supplied with district steam. If two buildings were to be selected where widely varying corrosion difficulties had been encountered, thorough and precise research, such as that of Professer Seeber and his co-workers, would disclose the variables accounting for satisfactory and unsatisfactory performance.

E. W. GUERNSEY[13] (WRITTEN): I question whether the conclusion as to the pro-

[12] Pres., Ferro-Nil Corp., New York, N. Y.
[13] Asst. Dir. of Research, Consolidated Gas, Electric Light and Power Co., Baltimore, Md.

portionality of the corrosion rate and oxygen concentration as shown in Figs. 5 and 6 can properly be drawn from the data given.

The individual points for the curves 6 and 7 are obtained from the slope of curves of the type of 4 and 5. The slope of curves 4 and 5 does not approach zero, as zero oxygen content is approached. In contradiction to this, the curves 6 and 7 are drawn so as to indicate a corrosion rate of zero, or at any rate, a very low figure for oxygen.

Even if curves 6 and 7 were correctly obtained from curves of the types 4 and 5, it would still not be clear that the results have to be interpreted as showing a relationship between oxygen and corrosion rate. As you will note from curves 4 and 5, oxygen concentration continually decreased during a run so that the observations at relatively low oxygen concentrations were always obtained after the specimen was already corroded. It is, therefore, not clear to what extent reduction in corrosion rate observed during the progress of an experiment should be attributed to decreasing oxygen concentration, as against the building up of a protective layer of corrosion products on the surface of the specimen.

In Memoriam

Name	Joined the Society	Died
D. M. Quay (*Charter and Presidential Member*)	1895	Feb. 1937
Samuel S. Clarke	1909	Feb. 1937
William J. Quigley	1920	Feb. 1937
Otto Marum	1931	Feb. 1937
Edgar W. Mandeville	1914	Mar. 1937
Champlain L. Riley (*Presidential Member*)	1906	Apr. 1937
Ralph T. Coe	1917	Apr. 1937
John H. Vogt	1925	Apr. 1937
O. P. Hood (*Honorary Member*)	1929	Apr. 1937
James F. Firestone	1914	May 1937
W. H. Wilson	1932	May 1937
R. Frank Eagar	1922	June 1937
George F. Stone	1918	July 1937
Ralph B. Hayward	1909	Aug. 1937
C. E. Dobbs	1921	Aug. 1937
J. E. McGinness	1903	Sept. 1937
William F. Gilling	1919	Sept. 1937
Frank B. Howell	1920	Oct. 1937
Earl L. Beard	1934	Oct. 1937
Hans G. Kreissl	1925	Nov. 1937
P. E. Fansler	1927	Nov. 1937
Robert A. Gemmill	1935	Nov. 1937
R. H. Martin	1937	Nov. 1937
W. B. Clarkson (*Life Member*)	1919	Dec. 1937

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